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FUEL EVALUATION FOR SMALL DIESEL ENGINES

INTERIM REPORT
BFLRF No. 274

By

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Southwest Research Institute (SwRI) conducted a project in support of the Auxiliary Powered Environmental Control System (APECS) for the Armored System Modernization Program. The purpose of this project was to develop concepts for an auxiliary power unit (APU) based on a small internal combustion engine that operates on heavy fuels such as diesel, JP-5, and JP-8.

After analyzing a comprehensive engine database, no diesel engines were found in current production that will meet the project targets; therefore, three approaches were identified as potential strategies to meet the project requirements: 1) Increase power output of small existing diesel engine; 2) Convert gasoline engine to spark-assisted diesel operation; 3) Design and develop new engine configurations.

A small, four-stroke diesel engine was located that will fit inside the available package volume with some modifications. A two-stroke gasoline engine was also located that will fit into the available space. Both engines will require major modifications to meet the power and fuel economy requirements on heavy fuels.

From a purely technical standpoint, the most promising approach appears to be the design and development of a lightweight diesel engine. However, the cost of this approach will be much greater than the two approaches described above. Therefore, our recommendation is that an existing diesel or gasoline engine be modified for demonstration purposes. After successful demonstration of the concept, a dedicated, lightweight diesel engine should be designed for this application.

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EXECUTIVE SUMMARY

Problems and Objectives: Operation of the main vehicle engine on armored tanks during night watch conditions results in very high fuel consumption. The purpose of this project was to develop concepts for an auxiliary power unit (APU) based on a small internal combustion (IC) engine that operates on heavy fuels such as diesel, JP-5, and JP-8. The ultimate goal of the APU is to achieve a substantial fuel savings by eliminating the need to run the main vehicle engine during night watch conditions.

Importance of Project: Small turbine engines are being considered for APUs due to their high power density and light weight. However, in the 75- to 100-hp range, it is well known that diesel engines have the potential to dramatically reduce fuel consumption, compared with turbine engines. This report concludes that a small IC engine can be developed for this application and quantifies the fact that small turbine engines cannot compete with diesel engines for this application when fuel consumption is taken into account.

Technical Approach: A database was used to identify the current diesel engine state-of-the-art in the range of 20- to 250-hp. New engine concepts were developed for achieving the performance and packaging goals. Each of the new concepts was compared with the others in terms of technical risk, development cost, and overall feasibility. Cost comparisons were also made between small turbine engines for this application and the new engine configurations which were recommended.

Accomplishments: This report concludes that there are no diesel engines available to meet the project goals; therefore, three approaches were identified as potential strategies to meet the project requirements:

1. Increase the power output of small existing diesel engine
2. Convert gasoline engine to spark-assisted diesel operation
3. Design and develop new engine configurations

A small, four-stroke diesel engine was located that will fit inside the required package volume with some modifications. This engine could be developed to provide the needed power output with some sacrifice in engine life. A two-stroke gasoline engine was also located that will fit into the available space. This engine will require that the combustion system be converted to spark-assisted diesel operation due to the low compression ratio. This engine will have a moderate life expectancy and higher fuel consumption than the project goals.

From a technical standpoint, the most promising approach appears to be the design and development of a lightweight diesel engine. This engine should come closest to meeting all requirements for this application. However, the cost of this approach will be much greater than the two approaches described above. Therefore, our recommendation is that an existing diesel or gasoline engine be modified to satisfy as many project targets as possible for demonstration

purposes. After successful demonstration of the concept of an IC engine for this application, a dedicated, lightweight diesel engine should be designed for this application.

Military Impact: The use of an internal combustion engine as an auxiliary power unit during night watch conditions will reduce the operating costs and improve the battle readiness of the Army's tanks. Since tanks spend the majority of their time under night watch or silent watch conditions, reduced fuel consumption with the small diesel engine will result in dramatic fuel cost savings. But more importantly, the reduced fuel consumption during night watch conditions will leave substantially more fuel on-board for battle conditions.

FOREWORD/ACKNOWLEDGMENTS

This work was performed by the Engine, Fuel, and Vehicle Research Division (EFVRD) for the Belvoir Fuels and Lubricants Research Facility (BFLRF) at Southwest Research Institute (SwRI), San Antonio, TX, under Contract No. DAAK70-87-C-0043, WD 34 for the peroid 1 May 1991 through 13 September 1991. Work was funded by the U.S. Army Belvoir Research, Development and Engineering Center (Belvoir RDE Center), Fort Belvoir, VA, with Mr. T. C. Bowen (STRBE-VF) serving as contracting officer's representative. Project technical monitor was Mr. Samuel J. Cerami (STRBE-FED).

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I. INTRODUCTION

A large portion of the fuel consumed in armored vehicle operations is during night watch conditions. In vehicles that use a gas turbine engine for motive power, the problem is aggravated by very poor specific fuel consumption characteristics at light load.

An internal combustion engine operating on the diesel cycle is inherently more efficient than a turbine engine, particularly in the range of power required for the Auxiliary Power Unit (APU) application (75- and 100-bhp). Internal combustion engines, however, present a challenge with respect to power density. Due to the extremely small space available for the APU, the size of the engine is severely limited. This size limitation means that the engine must possess very high specific output in order to meet the power demand. Because of the high specific output, durability also becomes an issue.

During the course of this project, data on existing diesel engines were analyzed, new combustion schemes were investigated, and potential concepts for meeting the project goals were developed.

II. TECHNICAL DISCUSSION

The objective of this project was to develop engine concepts that will produce 75- and 100-horsepower when operated on heavy fuel for auxiliary powered environmental control systems in armored vehicles. The specified performance and design goals for the auxiliary power unit (APU) are shown in TABLE 1.

The weight and size goals shown in TABLE 1 are for the complete auxiliary power unit, including the engine, two generators, a compressor, and heat exchangers. The best available information regarding the size and weight of the compressor, generators, and heat exchangers was subtracted from the total package specifications in order to determine the engine specific goals. The calculation of allowable engine size and packaging volume is shown in TABLE 2, and the resulting goals for the engines are shown in TABLE 3.

TABLE 1. Design Goals	
Characteristic	**Goal**
Size	17 in. x 22 in. x 31 in. (max)
Weight	275 pounds (max)
Fuel Consumption	15 kilograms per hour (max load)
Noise	70 dB-A @ 7 meters
Duty Cycles	1 - 100% for 10 kW electric power 2 - 100% for 10 kW electric power and 200 SCFM* of compressed air at 45 PSIG * - 400 SCFM for 100-horsepower engine
Endurance	3,000 hours before overhaul

After the engine specific performance requirements were established, several tasks were performed to determine the feasibility of meeting these goals with a heavy fuel internal combustion engine. First, a detailed study was performed to determine if any commercial engine was available that would meet the target specifications for this project. Second, the spark-assisted diesel combustion process was considered as a low-cost approach to developing an engine that would meet the target goals. Finally, three potential internal combustion engine options were evaluated (based on 100-hp case) in terms of cost, technical risk, and overall technical feasibility. The costs of these approaches were also compared with small turbine engine packages. Finally, the combustion methodology for the different approaches was addressed.

TABLE 2. Calculation of Allowable Engine Weight and Size

	Weight (lb)	Volume (ft^3)
Total	275	6.71
Generator	43	0.31
Alternator	31	0.41
Compressor	50	2.06
Heat Exchangers	30	0.88
Engine	121	3.05

TABLE 3. Calculated Performance Requirements

INDEX	75 hp	100 hp
Weight Specific Output (bhp/lb)	0.62	0.83
Volume Specific Output (hp/cu ft)	21.43	32.79
Brake Specific Fuel Consumption (lb/bhp-hr)	0.441	0.331

The following sections describe the results of each of these tasks.

A. Today's State-of-the-Art Diesel

Specifications were obtained for a wide range of diesel engines ranging from 20 horsepower to approximately 250 horsepower to identify the limitations of today's engine designs. These engine specifications are included in the Appendix. Some of the engine specifications that were investigated are listed below:

Rated Power	Rated Speed
Maximum Torque	Weight
Package Volume (L x W x H)	BSFC at Rated Power
Engine Cycle (2 or 4 Stroke)	Combustion System (DI, IDI, SPAD)
Aspiration (NA, T, TA, etc)	Cooling System (water, air)

Most of these parameters were plotted against rated power output to determine if today's state-of-the-art diesel production engine is capable of meeting the performance targets for this application.

1. Power to Package Volume Ratio

The most important design criteria for the APU application is the power to package volume ratio. The engine must fit into the available space before any other design targets can be considered. Figs. 1, 2, 3, 4, and 6 were plotted with data taken from the Appendix and show the effects of engine cycle, cooling media, aspiration, combustion system, and configuration on volume specific

power output. In addition to the specifications obtained from the engine manufacturers, a new prototype spark-assisted diesel engine (square symbol) is also included in these figures. This engine, based on an existing Cuyuna gasoline two-stroke engine, is currently under development at SwRI. This engine is in the early prototype stage and is not representative of production engines. The volume specific output of the Cuyuna engine is limited by the need to package a relatively large expansion chamber required for high power output.

It is interesting to note that Fig. 1 does not show an advantage for two-stroke engines over four-stroke engines in terms of volume specific power output. In fact, this figure indicates that four-stroke engines are capable of developing the same amount of power from a smaller package volume across the complete power range of 20- to 250-horsepower. One explanation for this trend is that the two-stroke engines included in this figure (with the exception of the Cuyuna engine) are uniflow scavenged, which means that they still have the extra package volume associated with a valve train similar to four-stroke engines. Heat rejection also plays a role in limiting two-stroke engine power density since there is half as much time for heat dissipation between combustion events. Finally, the figure points out that no available engines in the 75- to 100-bhp range meet the volume specific targets of 21.4 and 32.8 bhp per cubic foot, respectively.

Fig. 2 points out two important trends. First, the large majority of today's diesel engines are water cooled. Second, air-cooled diesels are found primarily in the very low power output range where specific power density and unit cost are low. Liquid-cooled engines are increasingly common because they offer improved control over cylinder liner and combustion chamber temperatures, allowing higher specific power to be reliably developed. For this application, high power density is a priority; therefore, a liquid-cooled design should be considered as a requirement.

Fig. 3 shows the relatively minor effect of aspiration on volume specific power density. In general, lower power engines are naturally aspirated, medium power engines begin to use turbocharging without aftercooling, and high power engines use turbocharging with water-to-air

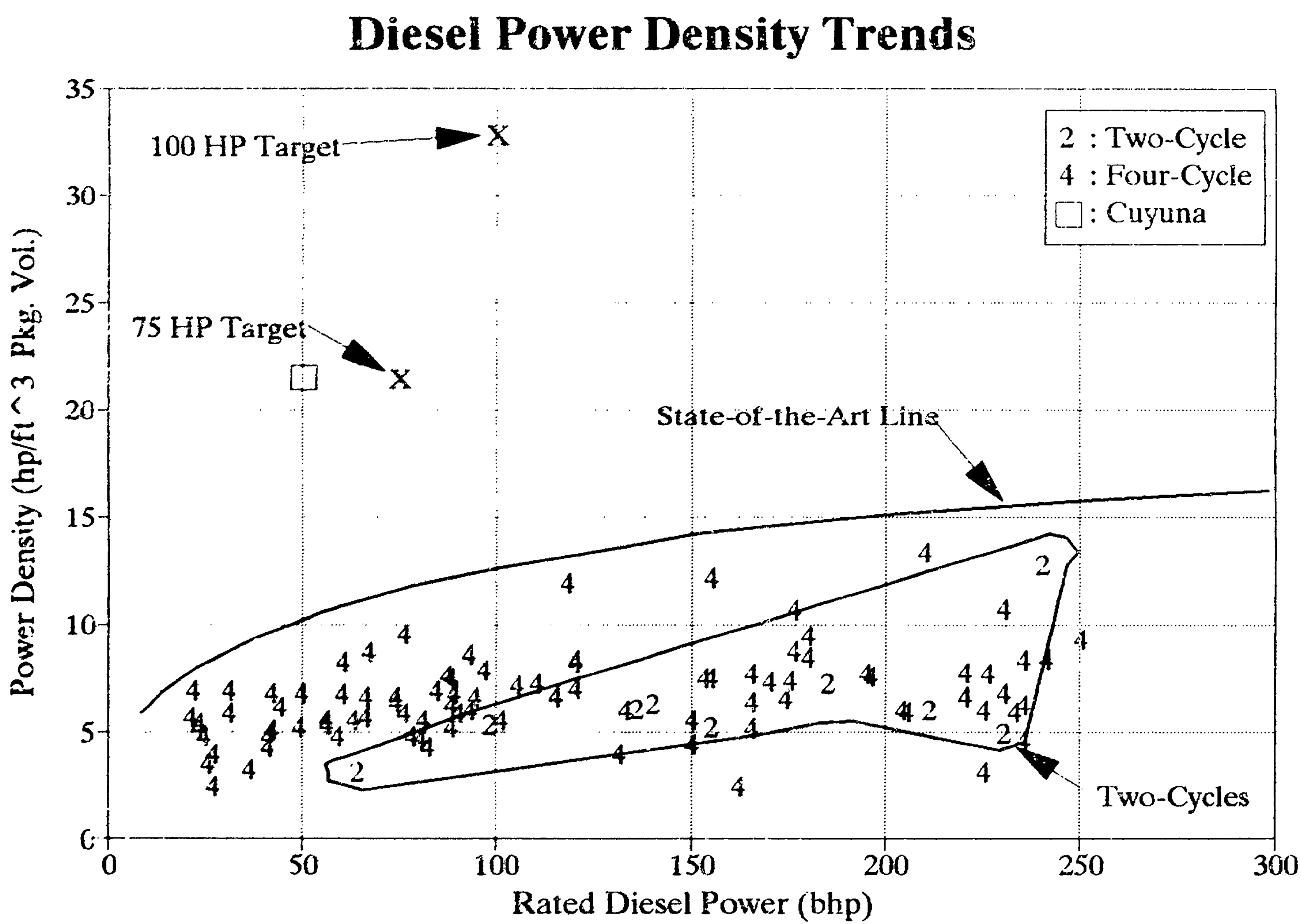
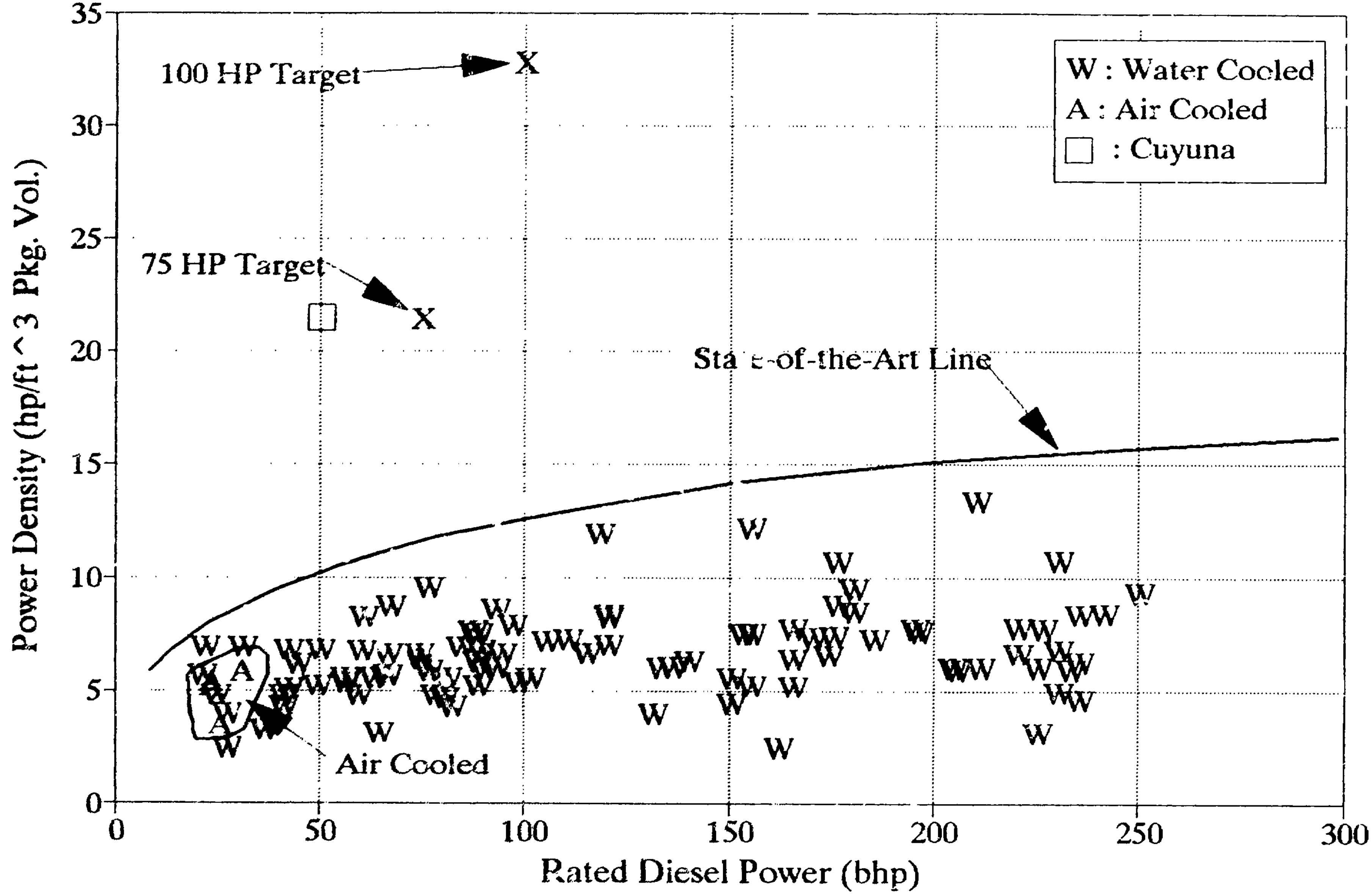


Figure 1. Relationship between engine cycle and volume specific power output

Diesel Power Density Trends



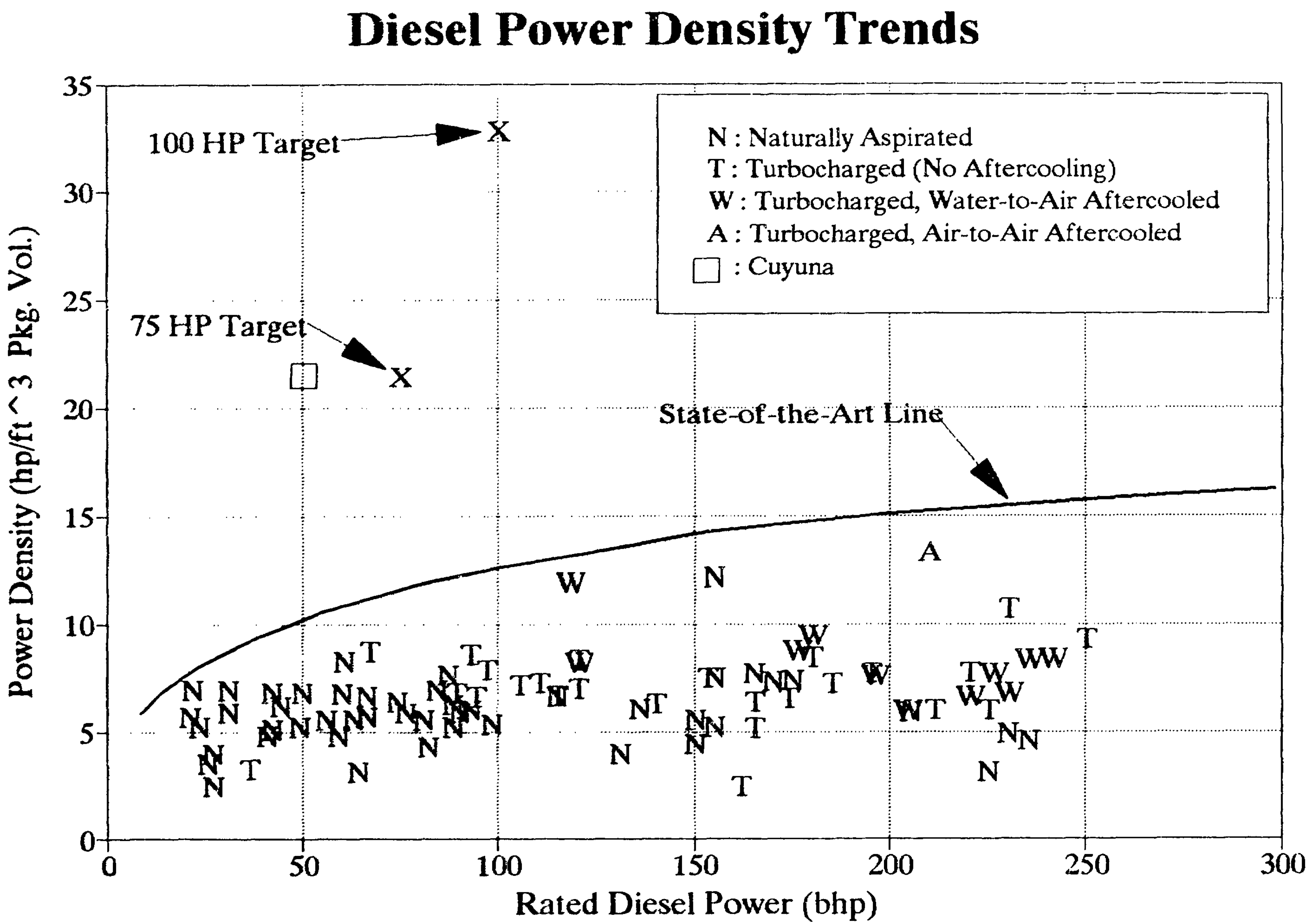


Figure 3. Relationship between aspiration and volume specific power output

or air-to-air aftercooling. Typically, the naturally aspirated engines develop acceptable power density by operating at a higher rated speed. Lower speeds are more readily achievable at the desired power output when turbocharging is employed. For the purposes of this application, turbocharging with water-to-air aftercooling will probably be necessary to meet the power density goals and reduce exhaust temperature and noise for strategic reasons.

Fig. 4 shows the combustion system types used on the diesel engines discussed in the previous figures. This figure indicates an advantage in power density for indirect injection (IDI) over direct injection (DI) in small diesel engines. The primary reason for increased power density with IDI diesels is improved air utilization.

Fig. 5 compares the Bosch smoke level in the exhaust stream of a typical IDI and DI diesel as a function of air/fuel ratio. At relatively rich air/fuel ratios, the IDI smoke level is much lower than the DI engine. This difference indicates higher air utilization in the IDI engine, which allows it to make more power by running richer.

In addition to increased air utilization, IDI combustion systems require a less sophisticated, less expensive fuel injection system. As a result, small engines tend to use IDI combustion systems more often, while DI combustion systems are found on larger diesel engines where fuel economy is more important than fuel injection system cost. The desired power density can be achieved with larger DI diesels by turbocharging and aftercooling. The topic of indirect versus direct injection will be discussed in more detail later, especially with regard to BSFC and fuel injection equipment.

Fig. 6 illustrates the effect of engine configuration on volume specific output. This figure shows that most engines in the 20- to 250-bhp range tend to have an in-line arrangement of cylinders. However, V-type engines and horizontally opposed engines typically have higher volume specific power density. Therefore, a V-type engine would probably be the most desirable configuration for this application, except that no diesel V-type engines exist in the desired power range.

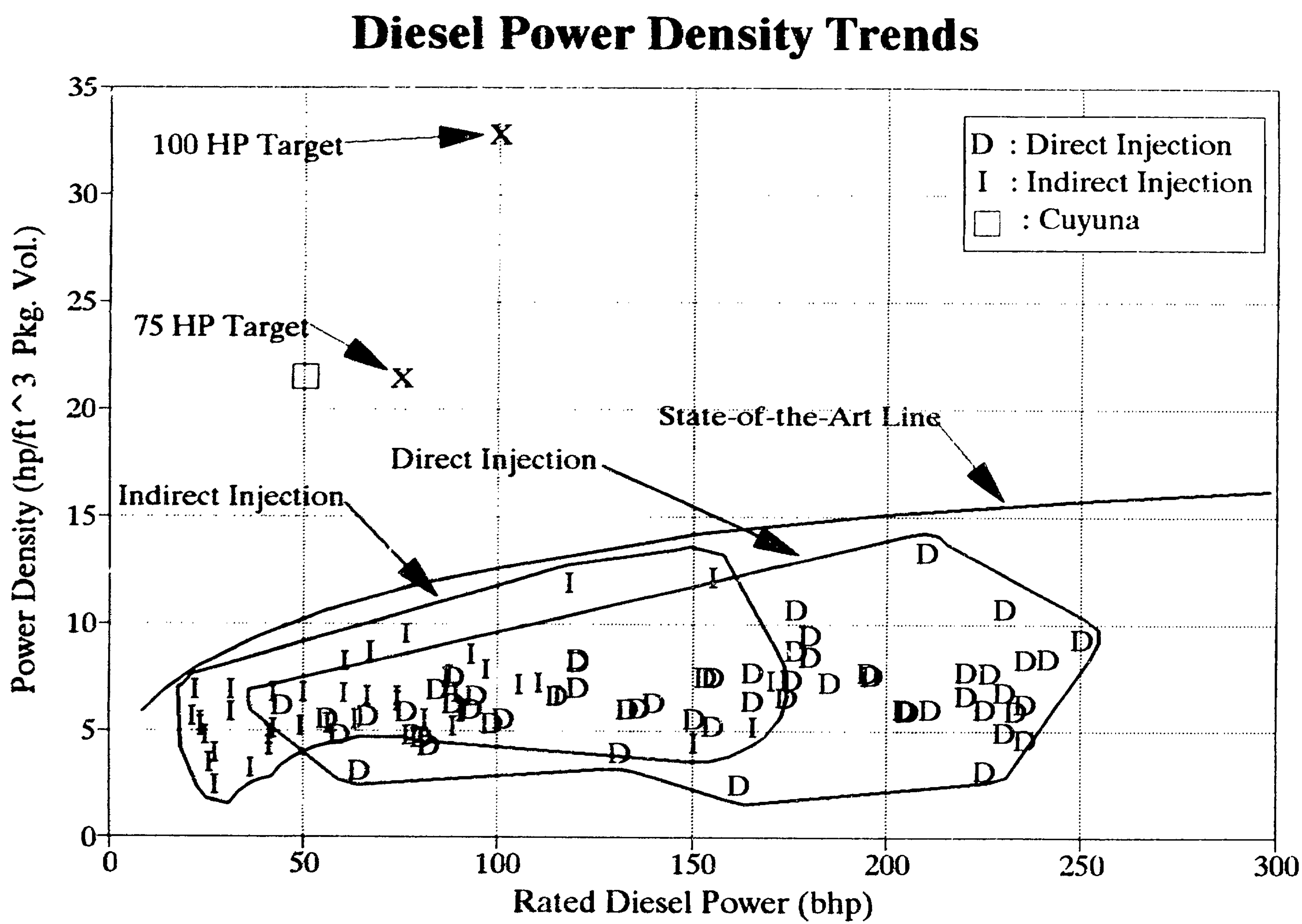
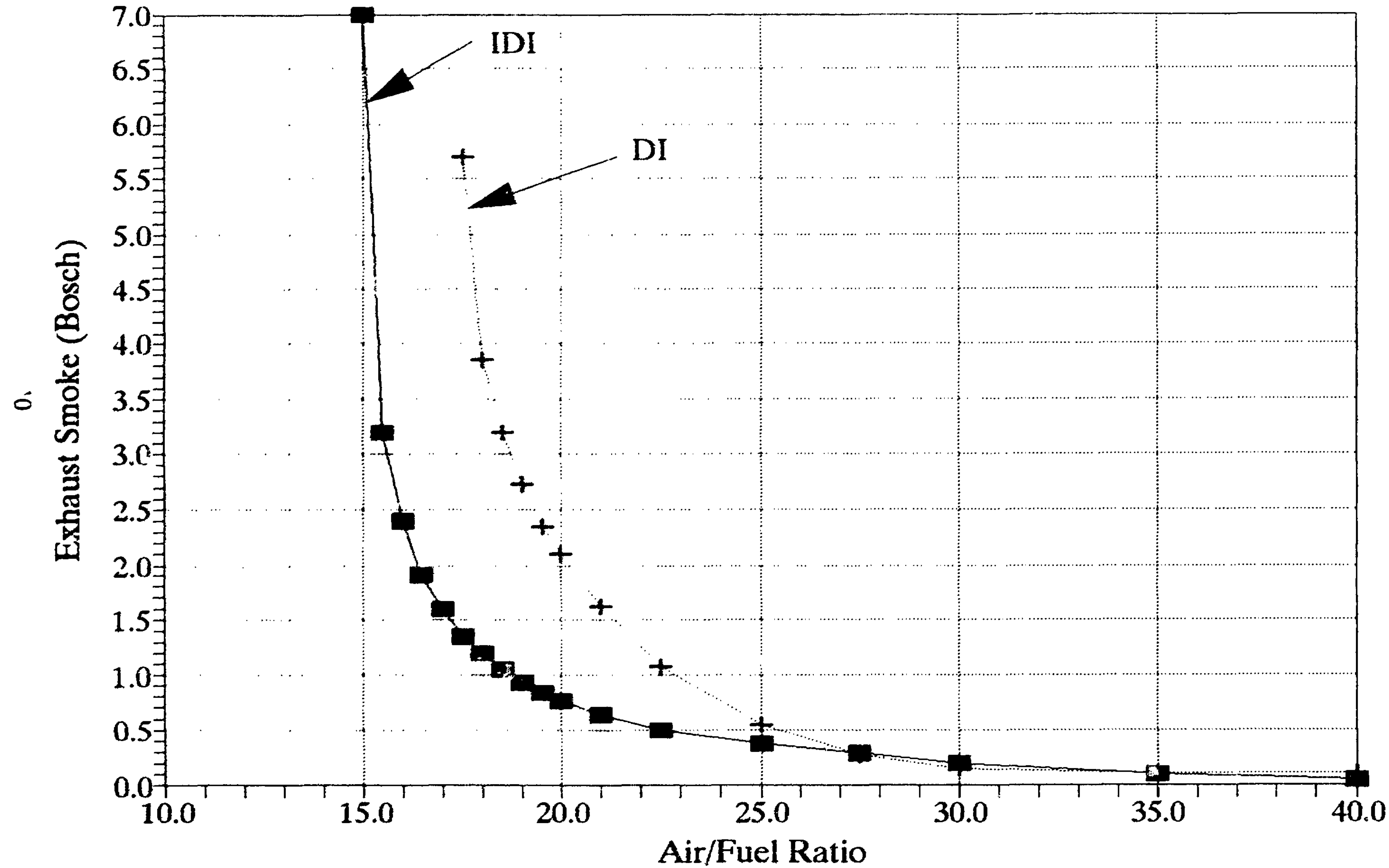


Figure 4. Relationship between diesel combustion systems and volume specific output

Typical Diesel Smoke Characteristics



Diesel Power Density Trends

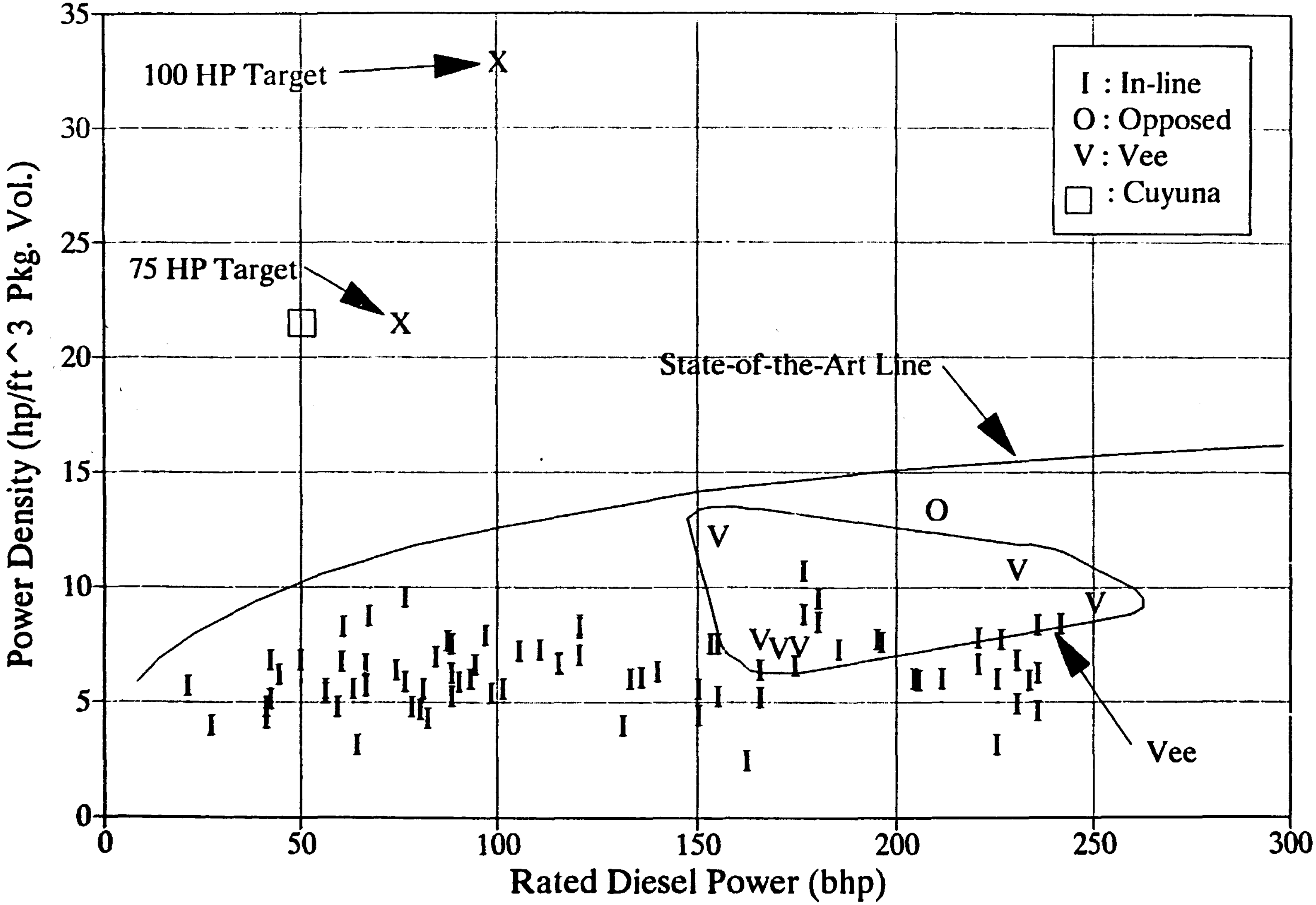


Figure 6. Relationship between engine configurations and volume specific power output

In summary, the power to volume ratio is perhaps the most important target for this application. Careful selection of design considerations such as engine cycle (piston ported two-stroke engine or four-stroke engine), configuration (in-line, V, opposed, etc.), and combustion system (DI or IDI) will play an important role in meeting this goal. As a result of this study, it appears that a liquid-cooled, piston-ported two-stroke engine, IDI diesel engine with a V configuration would come closest to meeting the packaging constraints and power output targets for this project.

2. Power to Weight Ratio

The power to weight ratio of the engine is also important for the auxiliary power unit (APU) application. As shown in the Appendix, high power density is a difficult goal for a diesel engine due to relatively low operating speeds and robust construction required to withstand the high compression ratio and firing pressures. Fig. 7 (derived from data in the Appendix) illustrates the shortfall of modem diesel engines from meeting the power to weight ratio targets at 75- and 100-bhp.

One of the most effective methods for increasing the power density of a given engine is to increase its rated speed while maintaining a given BMEP. For most diesel engines, the prospect of increasing the rated speed significantly is limited due to reciprocating component weights and fuel injection system limitations. As a result of these limitations, today's state-of-the-art 75- and 100-bhp diesel engines produce approximately 0.3-bhp/lb, compared with the targets for this application of 0.62- and 0.83-bhp/lb for the 75- and 100-bhp units, respectively.

Fig. 7 also includes data for two prototype engines that have been developed at SwRI. The triangular symbol shows a power to weight ratio of 0.4 at 45-bhp for a Sanshin two-stroke spark-assisted diesel engine. This engine has not been taken into production. The square symbol represents the target of an ongoing project at SwRI for the U. S. Navy to develop a Cuyuna two-cylinder, two-stroke engine to produce 50-bhp on diesel fuel, using a spark-assisted diesel combustion process. This engine is not in production either.

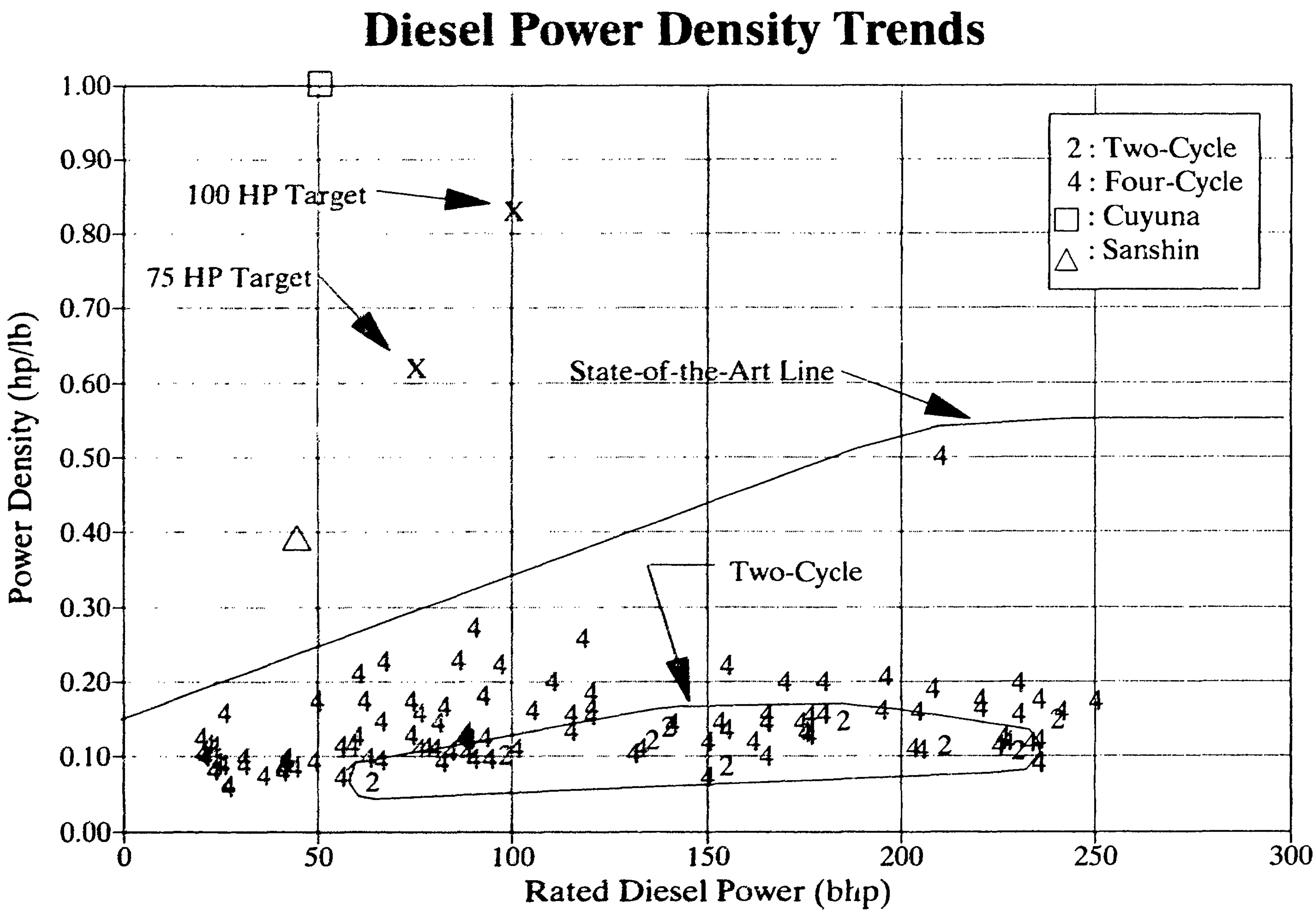


Figure 7. Relationship between engine cycle and weight specific power output

Since the engine speed and compression ratio can only be modified to a limited extent for existing diesel engines, there are two other approaches that should be investigated for reaching the desired power to weight ratio targets. These two approaches include advanced materials and selection of the correct combustion process. High strength to weight materials such as composites could be used for various structural components such as the crankcase, connecting rods, oil pan, intake manifold, and other components that are not exposed to the combustion chamber temperatures.

Using data from the Appendix, Fig. 8 depicts the advantage of using an indirect injection diesel combustion system compared with a direct injection combustion process. The IDI has the capability of providing significant improvements in the power to weight ratio of small diesel engines (i.e., 20- to 150-bhp). There are three primary reasons for the advantage of the IDI in this size range.

First, the IDI combustion chamber injects all of the diesel fuel into a small auxiliary combustion chamber where combustion begins. The peak combustion pressure inside this chamber is very high. As the combustion proceeds, the unburned fuel enters the main chamber at a very high temperature and velocity and mixes thoroughly with the remaining air while it burns. This injection strategy smooths out the combustion process in the main chamber and produces lower peak firing pressures on the piston crown which reduces the structural loading on the engine. This result allows the engine to be lighter in weight.

Second, the mixing of unburned fuel and air produced by the auxiliary chamber increases the air utilization compared with a similar DI diesel. Increased air utilization increases the power output. Thus, the IDI combustion system reduces engine weight by reducing peak firing pressure while simultaneously increasing power output by increasing air utilization. These two effects serve to increase the power to weight ratio noticeably.

Third, IDI engines can operate at higher engine speeds due to their simple fuel injection system requirements.

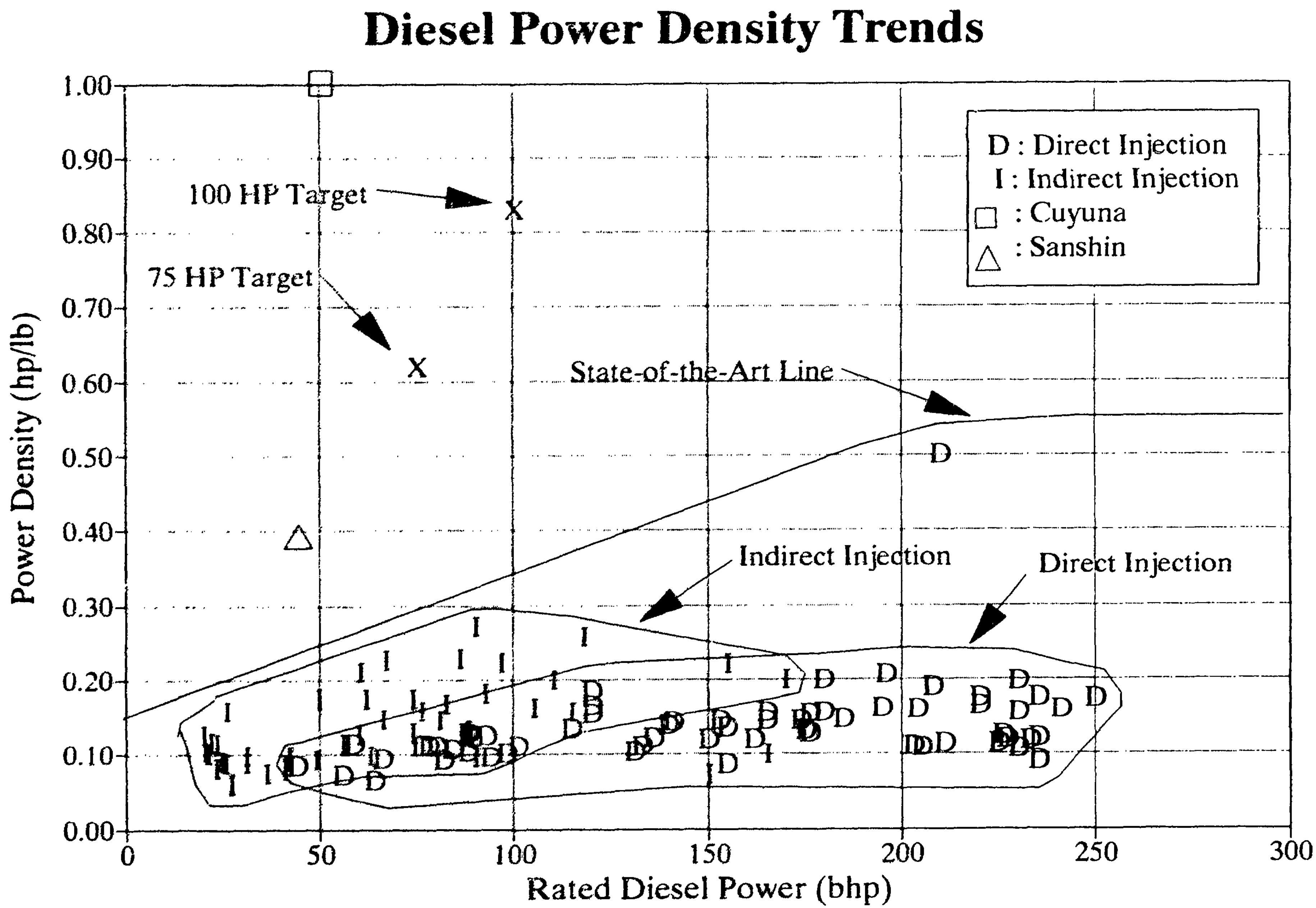


Figure 8. Relationship between diesel combustion systems and weight specific power output

Using data from the Appendix, Figs. 9 and 10 show the effect of cooling media and aspiration on the power to weight ratio. Fig. 9 shows very few diesel engines that use air-cooling, and very little benefit in terms of power density. Fig. 10 indicates a fairly consistent progression from naturally aspirated engines to turbocharged without aftercooling to turbocharged with water-to-air aftercooling and finally turbocharged with air-to-air aftercooling. This figure suggests the possibility of improving the power density of an existing engine simply by turbocharging and aftercooling. The primary disadvantage to this simplistic approach is increased mechanical loading on the engine.

3. BSFC

Using data from the Appendix, Fig. 11 illustrates one of the disadvantages of using an IDI combustion system compared with a DI combustion system. IDI engines have poor BSFC compared to DI engines because of increased heat rejection to the coolant and pumping losses through the auxiliary chamber nozzle. Fig. 11 indicates that the BSFC goal of 0.44 lb/hp-hr can be met with an IDI system for the 75-bhp engine, but the 0.33 goal for the 100-bhp engine will be very difficult to meet, even with the best of today's DI diesels. DI diesels are difficult to scale down below 100 horsepower because of the small quantity of fuel being injected and the short distance the fuel plume can travel before impinging the cylinder wall. If the power to volume and power to weight ratio targets are more important than the BSFC targets for this application, then IDI combustion systems should be considered as the best strategy for compression ignition combustion.

Fig. 12 illustrates the trade-off between engine weight and fuel consumption for the turbine, IDI, and DI cases. Even though the turbine engine is less than half the weight of the diesel engines, its high fuel consumption means that the total weight of the engine and fuel will be much higher for the turbine engine than the diesel engine, if more than two hours of operation at 100-hp are required. The break-even point, in terms of total engine and fuel weight, for the DI and IDI diesels is approximately eight or nine hours of operation at 100hp according to this figure.

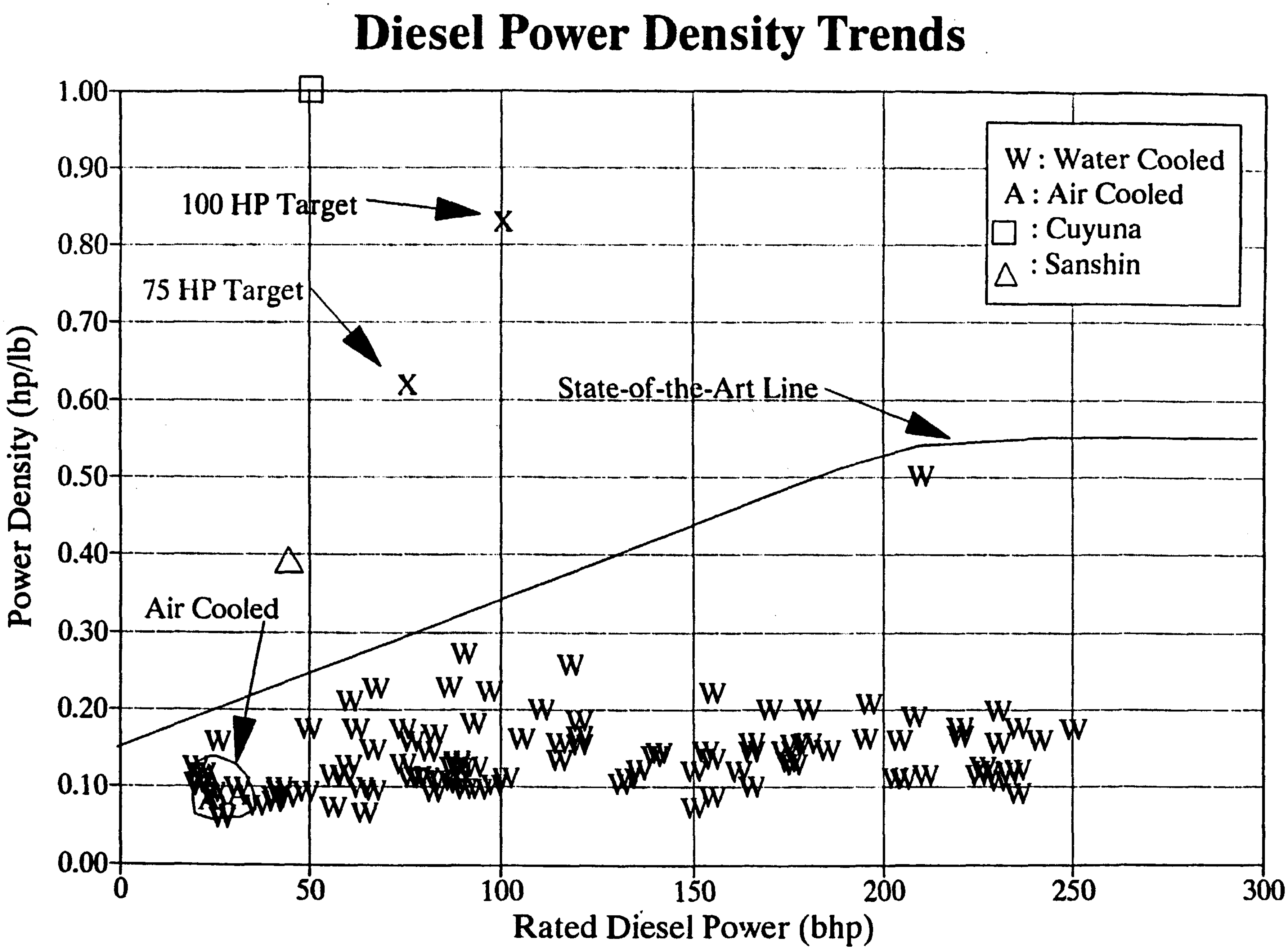
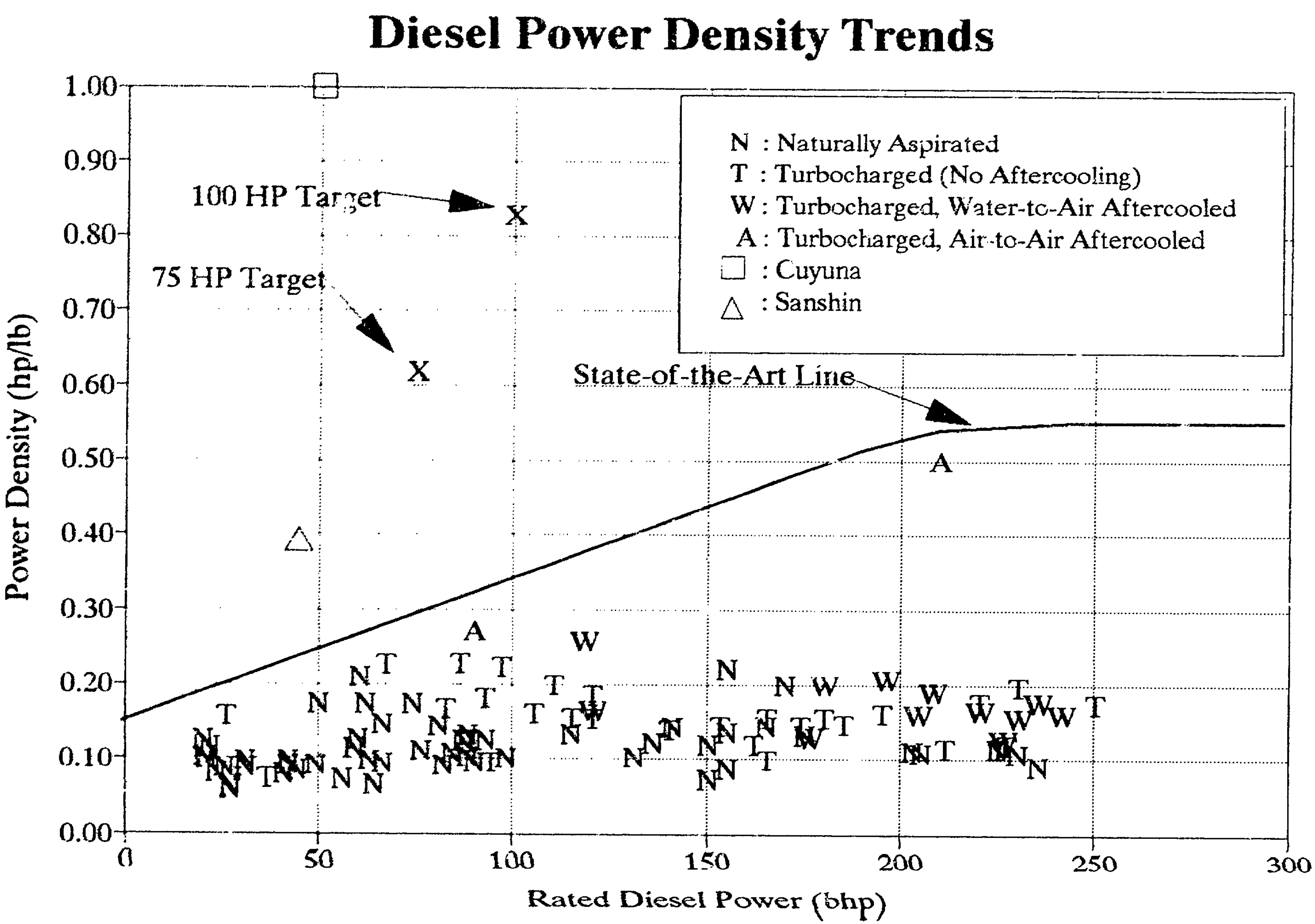


Figure 9. Relationship between cooling media and weight specific power output


Diesel Power Density Trends
Power Density (hp/lb)
Rated Diesel Power (bhp)
1.00
0.90
0.80
0.70
0.60
0.50
0.40
0.30
0.20
0.10
0.00
0
50
100
150
200
250
300
N : Naturally Aspirated
T : Turbocharged (No Aftercooling)
W : Turbocharged, Water-to-Air Aftercooled
A : Turbocharged, Air-to-Air Aftercooled
□ : Cuyuna
△ : Sanshin
100 HP Target
75 HP Target
State-of-the-Art Line

Diesel Fuel Consumption Trends

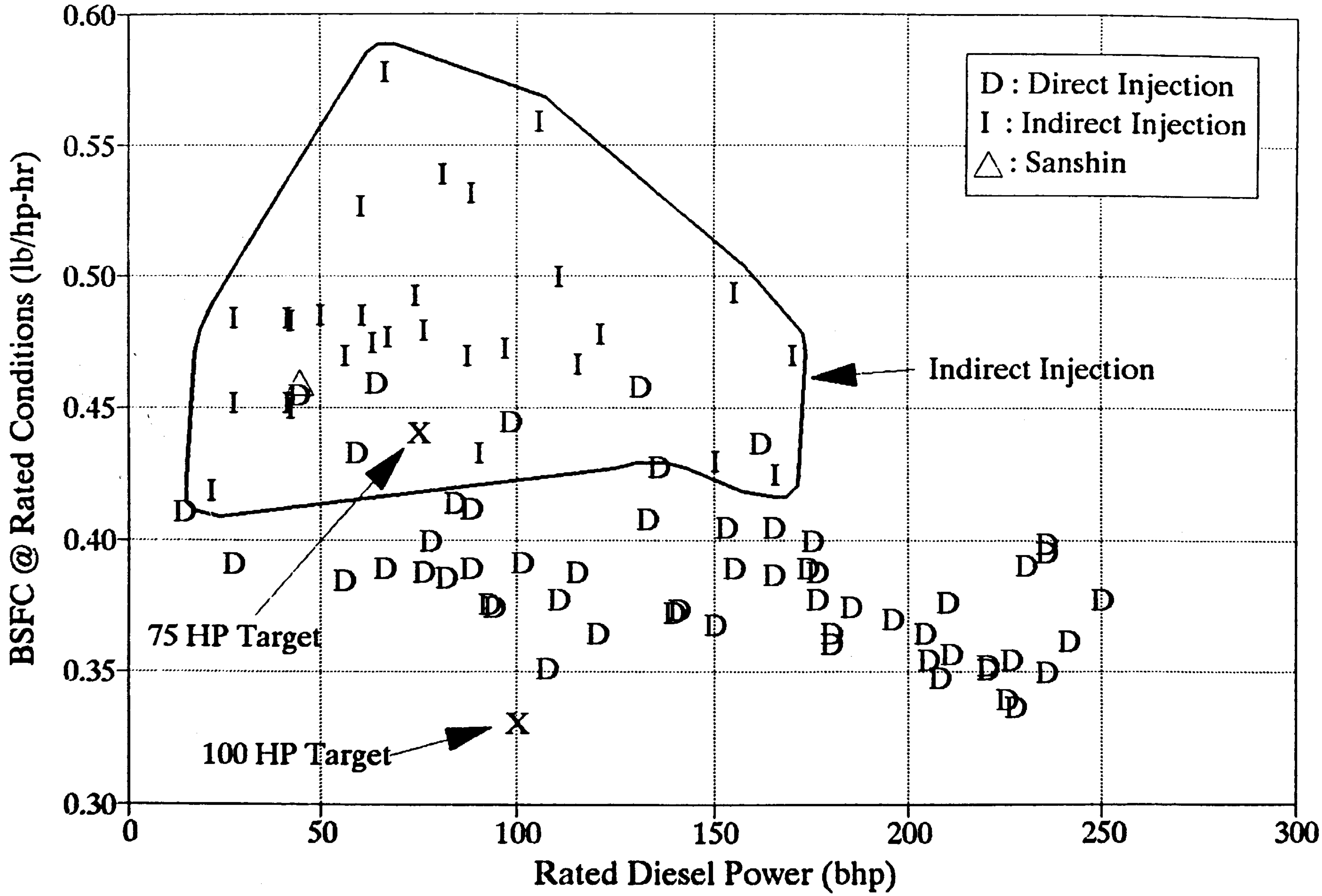
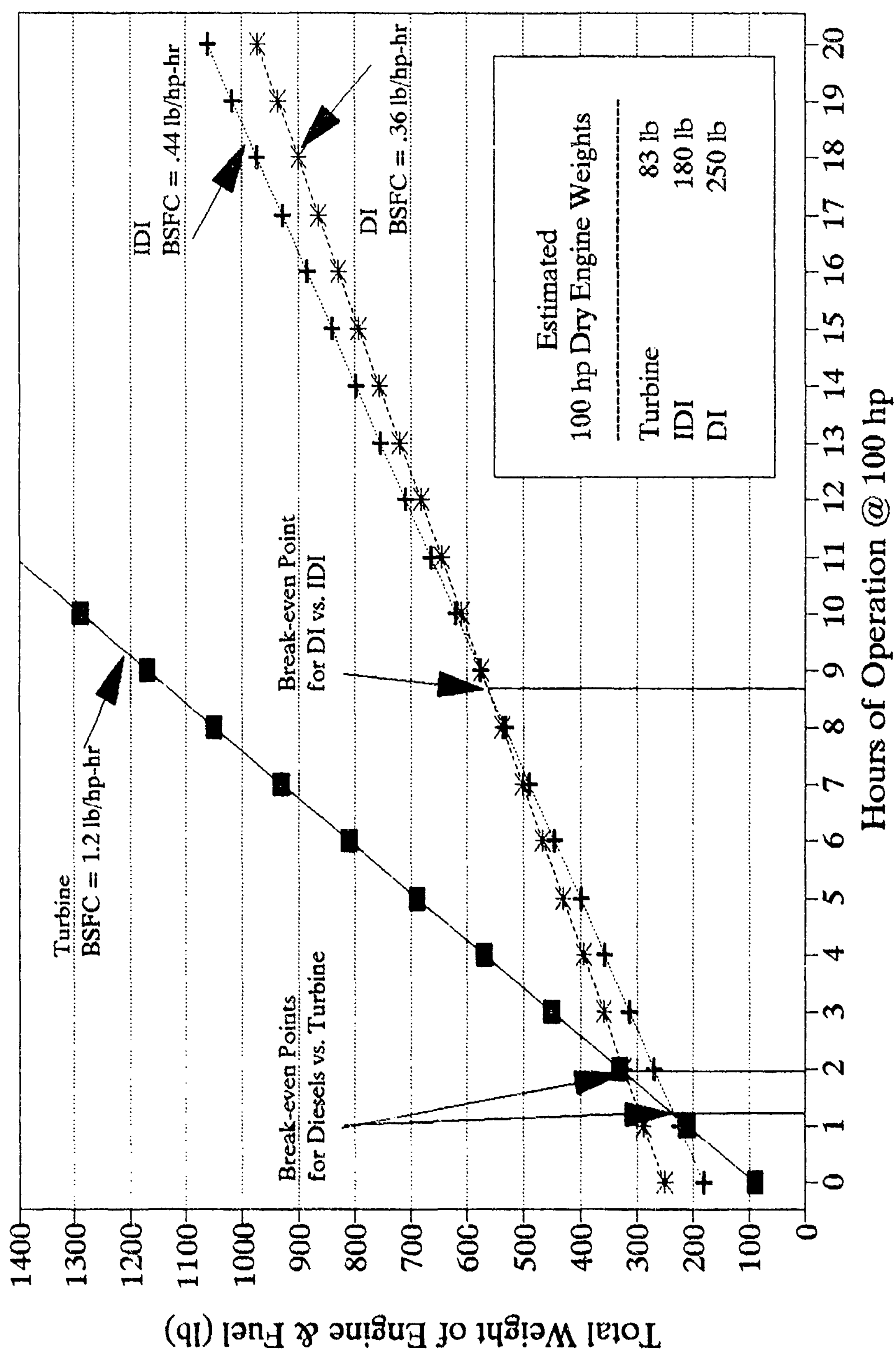


Figure 11. Comparison of bsfc levels for DI and IDI combustion systems

Effect of Fuel Consumption on Total Weight



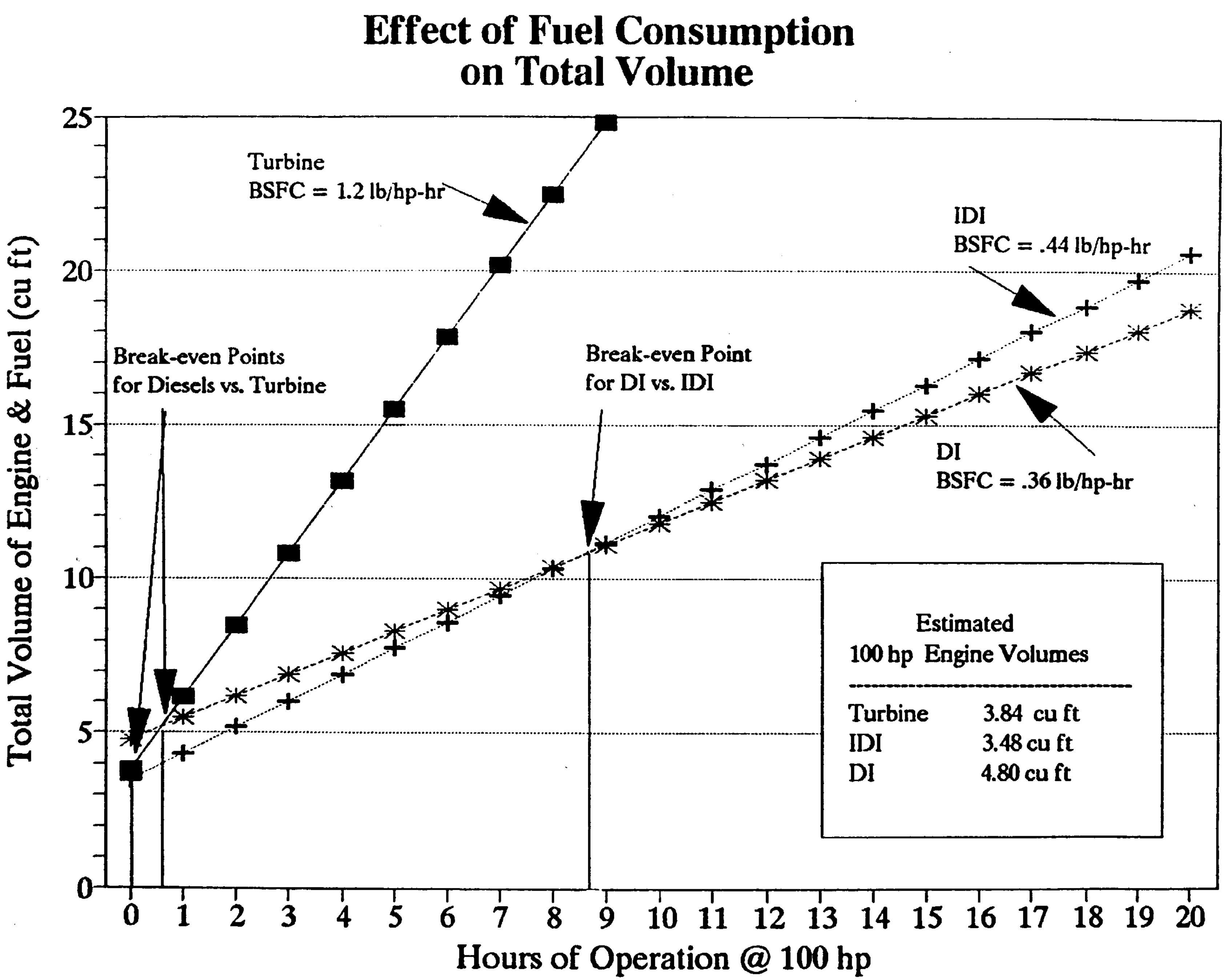


Figure 13. Trade-off between engine package volume and fuel consumption for turbine engine and IDI and DI diesel engines

Fig. 13 shows the same trade-offs expressed in terms of total engine and fuel package volume. These two figures clearly show that the turbine power plant is inferior to the diesel engines in terms of total system packaging. If the additional volume of the air filtration system for the turbine engine is considered, the turbine engine will look even less favorable compared with the diesel engines.

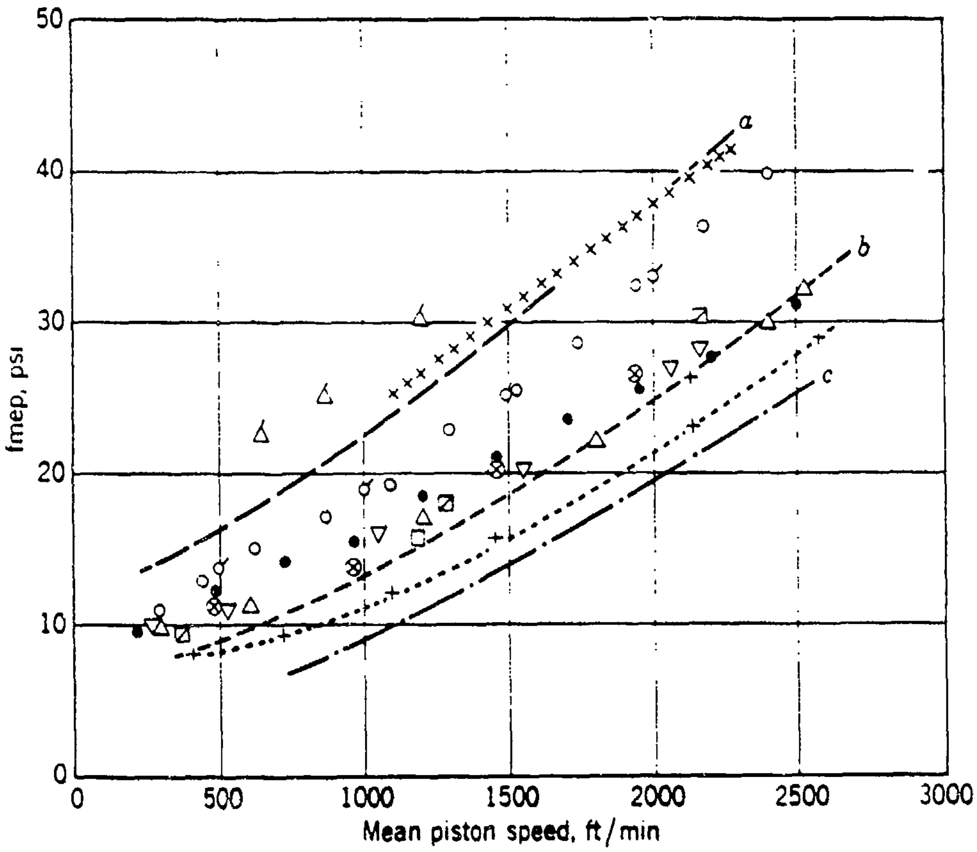
The only other disadvantage of the IDI diesel, aside from the higher fuel consumption, is cooling of the auxiliary chamber nozzle under very high load levels. Since the application being studied will require high power density, special materials (such as ceramics) or special cooling will probably be needed for the nozzle area and piston crown.

Additional difficulties in meeting the BSFC targets for this application deserve mention. In order to meet the power output, power to volume, and power to weight goals simultaneously, the engine will need to be small and operate at high speeds. As a general rule, small engines have lower thermal efficiency than larger engines due to the increased surface area to volume ratio of the combustion chamber. In addition, the friction power losses increase with engine speed (or piston speed) as shown in Fig. 14. Finally, it is difficult to scale fuel injection equipment down effectively, which translates into reduced air/fuel mixing and higher fuel consumption for smaller engines.

4. Heat Rejection

In general, the high power density required for this engine application could make the targets more difficult to meet with a two-stroke engine. Two-stroke engines fire twice as often as four-stroke engines at the same speed, leaving less time for heat transfer to take place.

Regardless of the engine cycle chosen, heat rejection at full load conditions will be one of the most difficult challenges for this engine application. The piston will probably be one of the most difficult components to cool. Therefore, advanced engine cooling strategies such as piston cooling with an oil jet could prove to be necessary. Insulated exhaust ports may also be needed to reduce the heat exchanger capacity requirements. If turbocharging is adopted, then the



Symbol		Year	Bore	Stroke	Reference
△	Off-highway Diesel	1942	4.5	5.5	9.730
a — —	U.S. passenger cars (max)	1938	—	—	1.10
b - - - -	U.S. passenger cars (min)	1938	—	—	1.10
c —·—	Aircraft engine, V-12	1943	5.5	6	9.703
○	U.S. passenger car, V-8	1949	3.81	3.38	9.57
⊗	U.S. passenger car, V-8	1952	3.81	3.38	9.710
△	U.S. passenger car, 6	1952	3.56	3.60	9.711
▽	U.S. passenger car, V-8	1951	3.50	3.10	9.712
●	U.S. passenger car, V-8	1952	3.38	3.25	9.713
□	Marine Diesel*	1952	11.8	17.7	9.731
♂	Automotive Diesel	1940	3.75	5.0	9.732

*From indicator cards – all others motoring.

+ ·······	U.S. passenger car	1946	3.25 x 4.38	9.57
⊠	U.S. passenger car	1946	3.19 x 3.75	9.57
× × × × ×	1.25" x 1.25" overhead-valve test engine			9.74
p_e/p_i	1.0 in all cases			

Source: C. F. Taylor, "The Internal Combustion Engine in Theory and Practice," Vol. 1, Page 350

Figure 14. Effect of engine (piston) speed on engine friction

maximum amount of charge cooling will be needed to control combustion chamber component temperatures.

B. Spark-Assisted Diesel Combustion

The spark-assisted diesel (SPAD) concept has been applied to different engines by SwRI in order to produce lightweight, high-speed diesel engines. The SPAD combustion system is necessary for these applications due to the high peak combustion pressures and rates of pressure rise that would be experienced with a true compression ignition diesel.

In a conventional compression-ignited diesel engine, the temperature and pressure of the combustion air are raised to the level at which the diesel fuel will begin to ignite soon after injection into the combustion chamber. The small delay between injection of the diesel fuel and ignition is called the ignition delay. For a typical direct-injected diesel engine with a compression ratio of 16:1, the ignition delay is very small as long as the cetane number is greater than 40. A typical cylinder pressure diagram is illustrated in Fig. 15. This figure shows the short ignition delay for a 16:1 compression ratio.

However, in order to build a lightweight diesel engine, the peak firing pressures and rates of pressure rise must be reduced in order to lower the structural load on the engine block, cylinder head, and bearings. If the compression ratio is dropped, the expected result would be a reduction in peak firing pressures and rates of pressure rise. However, the reduced temperature and pressure at the time of injection cause an increase in the ignition delay, which actually increases the peak pressure and pressure rise rate due to an increase in the premixed combustion. These trends are illustrated in Fig. 16, which begins with the 16:1 compression ratio described above. The injection timing and rate of injection are assumed to be constant for each compression ratio. This figure shows a more severe combustion condition at a 12:1 compression ratio than at the 16:1 compression ratio. This figure helps illustrate the counterproductive effect of lowering the compression ratio on engine performance and life due to the increased ignition delay.

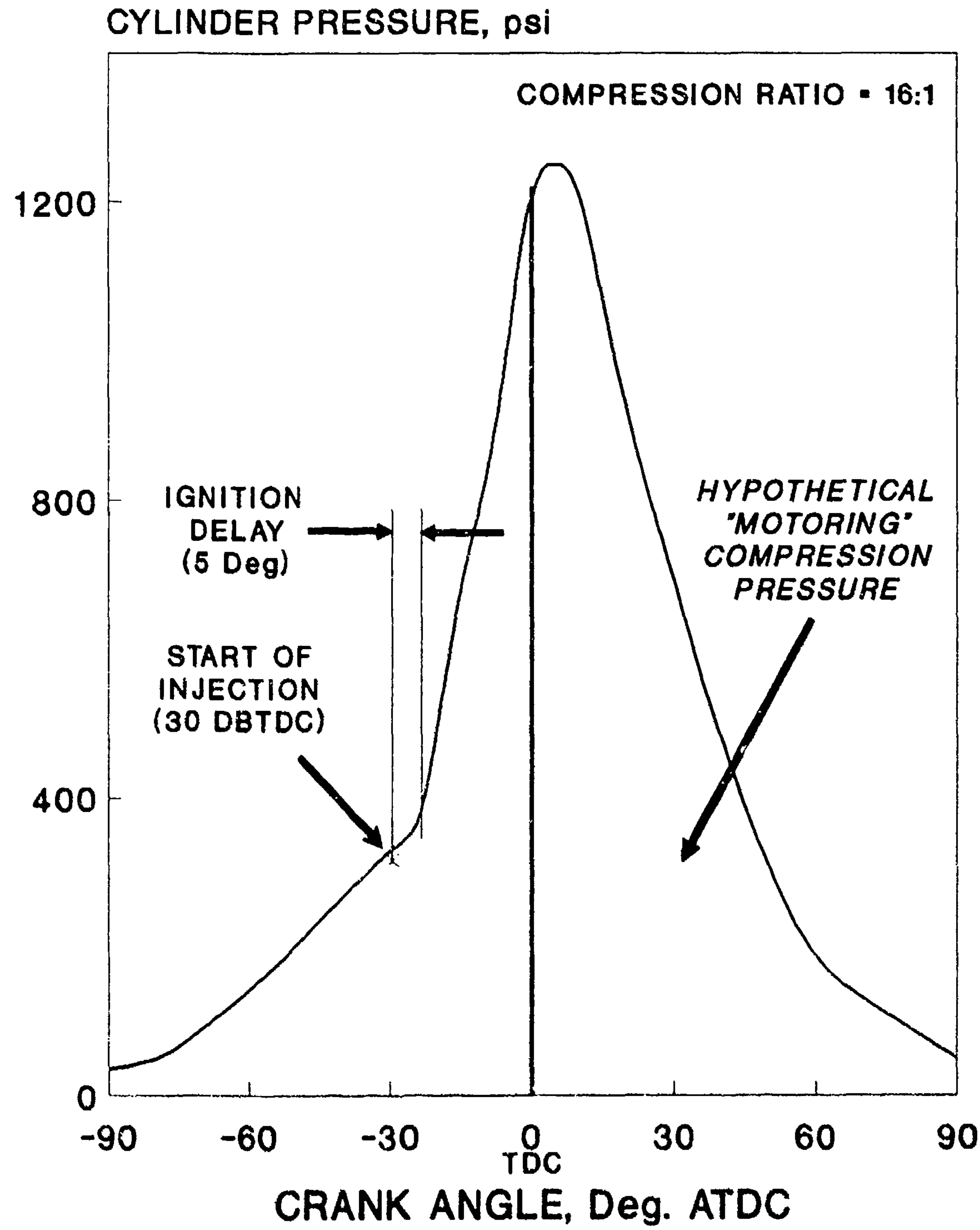


Figure 15. Typical cylinder pressure diagram for DI diesel engine with 16:1 compression ratio

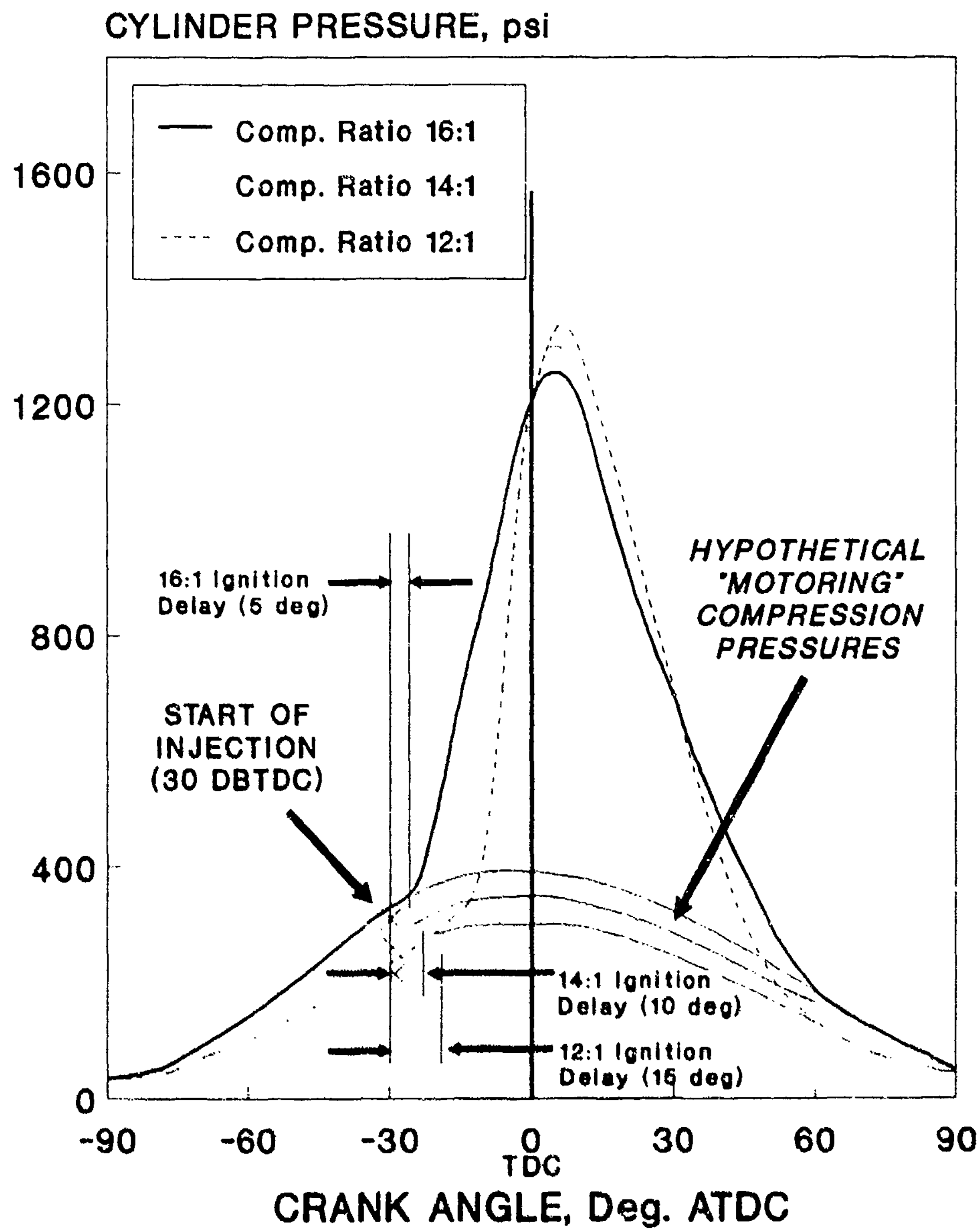


Figure 16. Effect of compression ratio on ignition delay for typical DI diesel engine

The spark-assisted diesel combustion system helps alleviate the disadvantages of increased ignition delays with the lower compression ratio. This strategy allows the lower compression ratio to be used in order to achieve lower peak firing pressures and rates of pressure rise without suffering from increased ignition delays. The spark-assisted diesel combustion system operates by striking the spark plug very soon after the point of injection. The spark energy ignites the fuel in the immediate vicinity of the plug and begins the combustion process much sooner than if the fuel's ignition delay were allowed to control the beginning of ignition. The effect of the spark-assisted diesel on the cylinder pressure diagram at a 12:1 compression ratio is illustrated in Fig. 17. This figure compares the normal compression ignition diesel combustion with the spark-assisted diesel combustion at 12:1 compression ratio with identical fuel injection events.

C. Diesel Concepts for APU's

Since no production diesel engines are available to meet the targets of this application, a new engine must be designed or developed. Three primary approaches have been identified as possibilities to meet the desired specifications:

1. Increase power output of small existing diesel engine
2. Convert an existing gasoline engine to spark-assisted diesel (SPAD) operation
3. Design and develop a lightweight diesel engine specifically for this application

Each of these approaches is discussed below.

1. Increase Power Output of Small Existing Diesel Engine

In order to modify an existing diesel engine for this application, an engine that fits into the available package volume must be chosen. A 21-hp Yanmar diesel engine was chosen for consideration of this approach. Specifications for this engine are shown in TABLE 4.

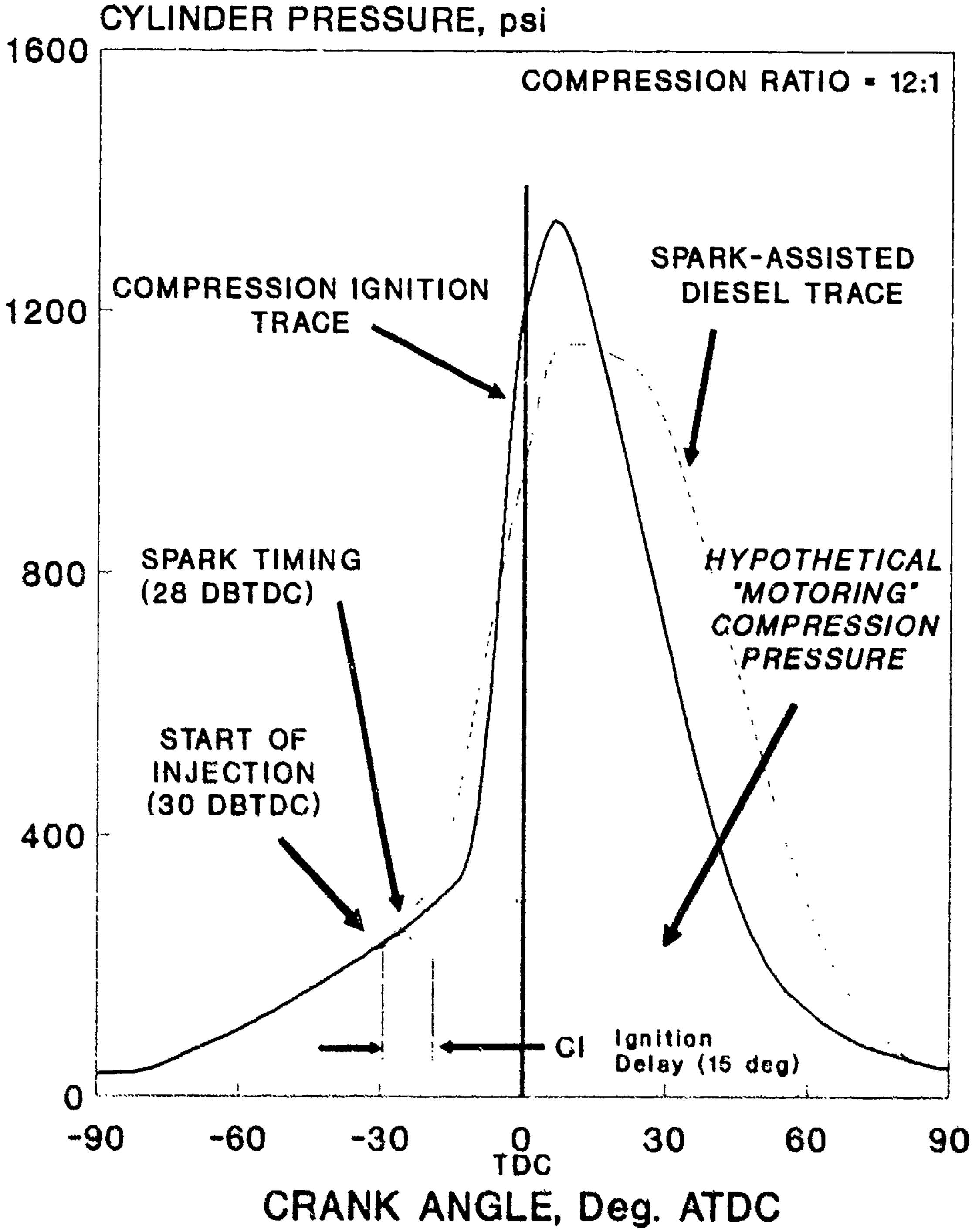


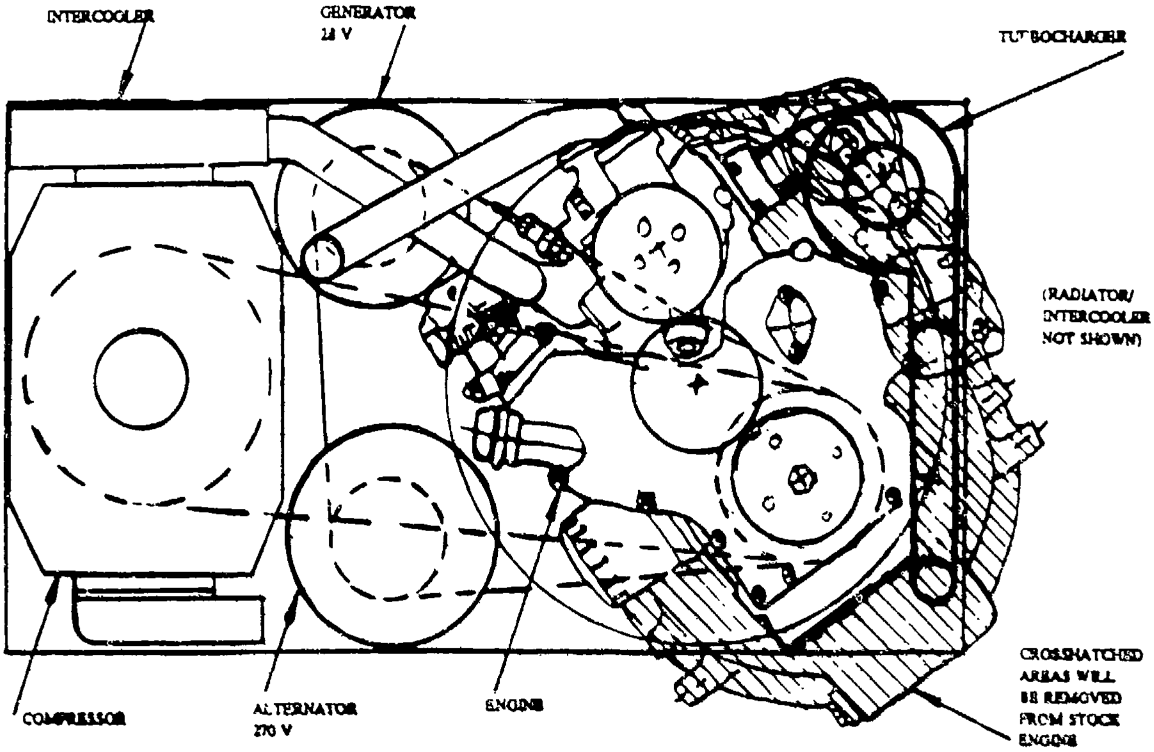
Figure 17. Illustration of combustion "smoothing" on low compression ratio diesel with spark-assist

TABLE 4. Yanmar Engine Specifications

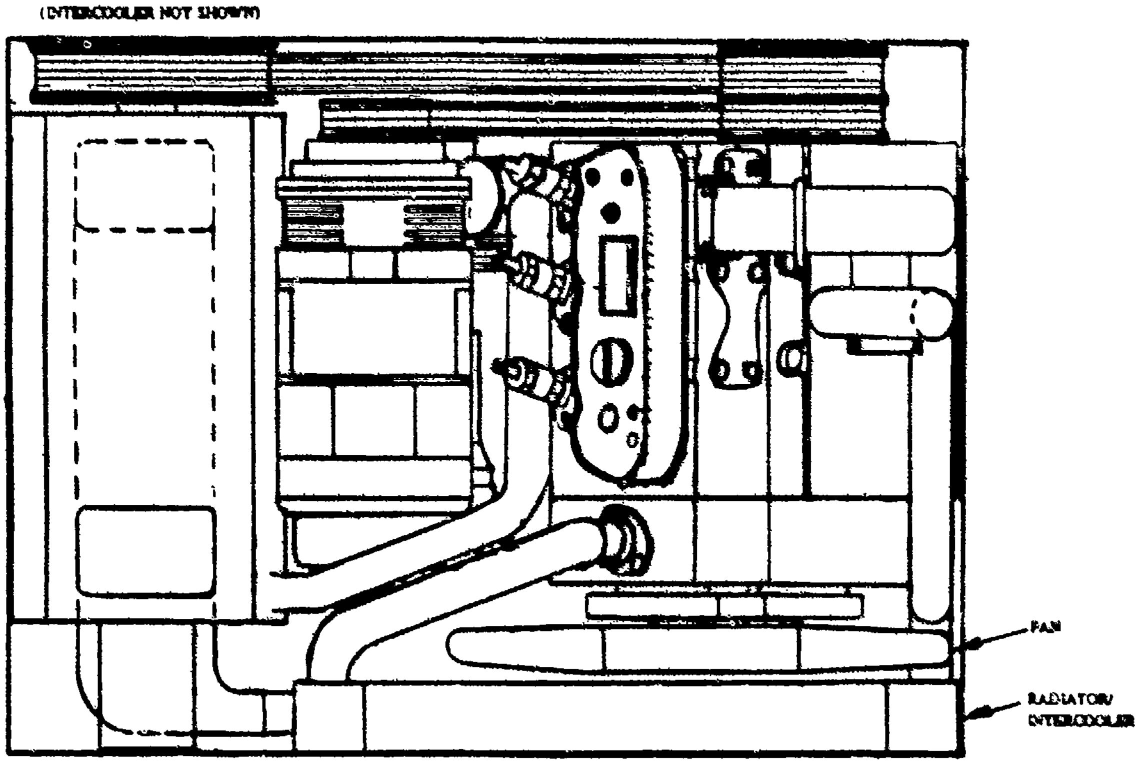
Manufacturer:	Yanmar
Model:	3TNA72E
Bore x Stroke:	72 x 72 mm
Displacement:	880 cc (53.6 cu in)
Rated Power:	21-hp at 3600 RPM
BMEP:	86 psi at 21-hp
Configuration:	In-line, three-cylinder
Cycle:	Four-Stroke
Induction:	Naturally Aspirated
Combustion:	Indirect Injection (IDI)
L x W x H:	18.5 x 17.4 x 19.8 in.
Dry Weight:	187 lb

As indicated by the engine specifications in TABLE 4, the Yanmar is nearly three inches taller than the package volume height shown in TABLE 1 and is heavier than the allowable engine weight calculated in TABLE 2. However, the engine height can be reduced significantly by converting the lubrication system to a dry sump arrangement with a remotely located oil reservoir. The engine can also be tilted about 15 or 20 degrees in order to use the diagonal box dimension to accommodate the engine height. Fig. 18 illustrates a possible packaging arrangement using the Yanmar engine in the allocated space.

Increasing the power output of the Yanmar engine from 21-hp to 75-hp and 100-hp will require supercharging and increasing the rated speed of the engine. It should be possible to increase the rated speed of the Yanmar to 5000 RPM. IDI fuel injection systems are available for four-stroke engines in this speed range, and several IDI engines in this power range are rated from 4000 to 4500 RPM as shown in Fig. 19 (Data derived from Appendix). Potential limitations to increasing the engine's rated speed from 3600 RPM to 5000 RPM may include valve train dynamics, rod and rod bolt stresses, piston pin stresses, and the bearing systems. Calculations will be required to determine the necessity for component design changes in order to accommodate the increased operating speed. Premium grade components should also help meet the life targets at the higher operating speeds. In fact, piston speed is often used as an indication of engine longevity, and Fig. 20 (taken from Appendix data) shows that the Yanmar's piston speed at 5000 RPM will still be lower than other production diesel engines.



END VIEW



TOP VIEW

Figure 18. Illustration of APU configuration using the Yanmar diesel engine

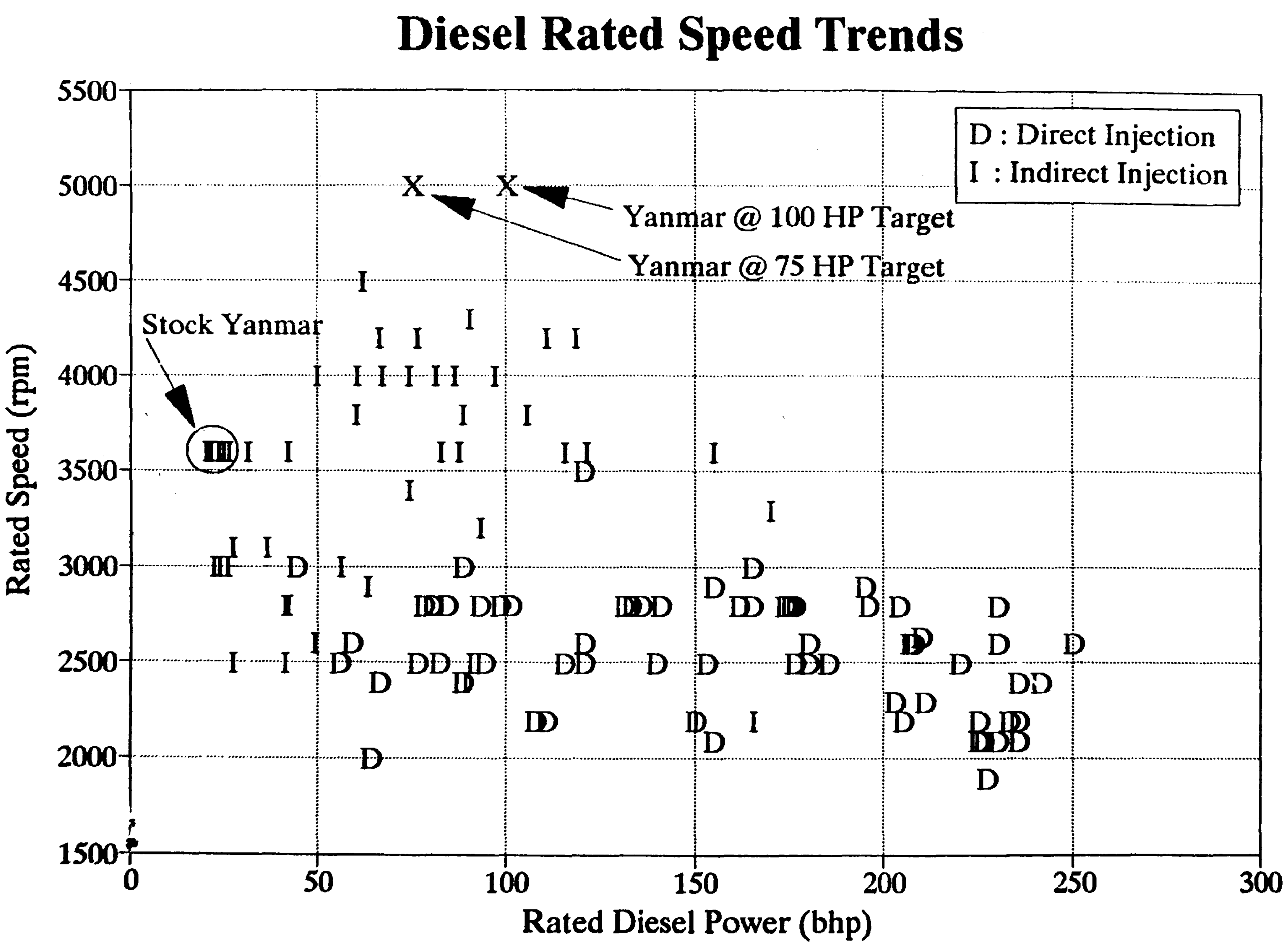
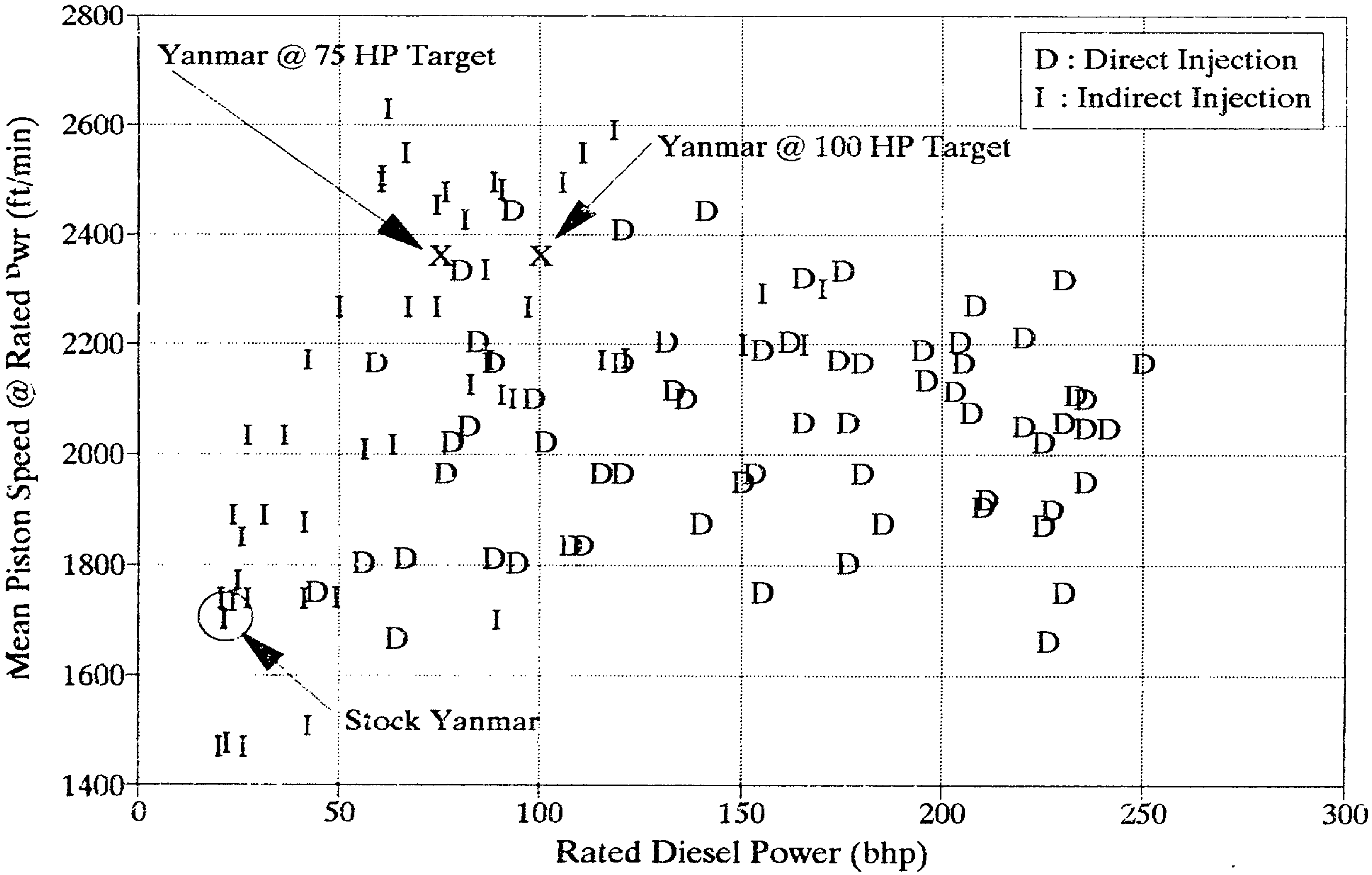


Figure 19. Rated speeds for wide range of small diesel engines

Diesel Piston Speed Trends



Another possible limitation to meeting the power requirements with the Yanmar engine is the engine structure. Supercharging the Yanmar sufficiently to produce 75- and 100-hp at 5000 RPM will increase the BMEP from 86 psi to 222 psi and 290 psi, respectively. These BMEP levels are very high compared with other production diesel engines as indicated in the Appendix and illustrated in Fig. 21. The increased BMEP will place heavy stresses on the cylinder heads, cylinders, crankshaft, bearings, and crankcase, unless the peak firing pressure is held constant by reducing the compression ratio.

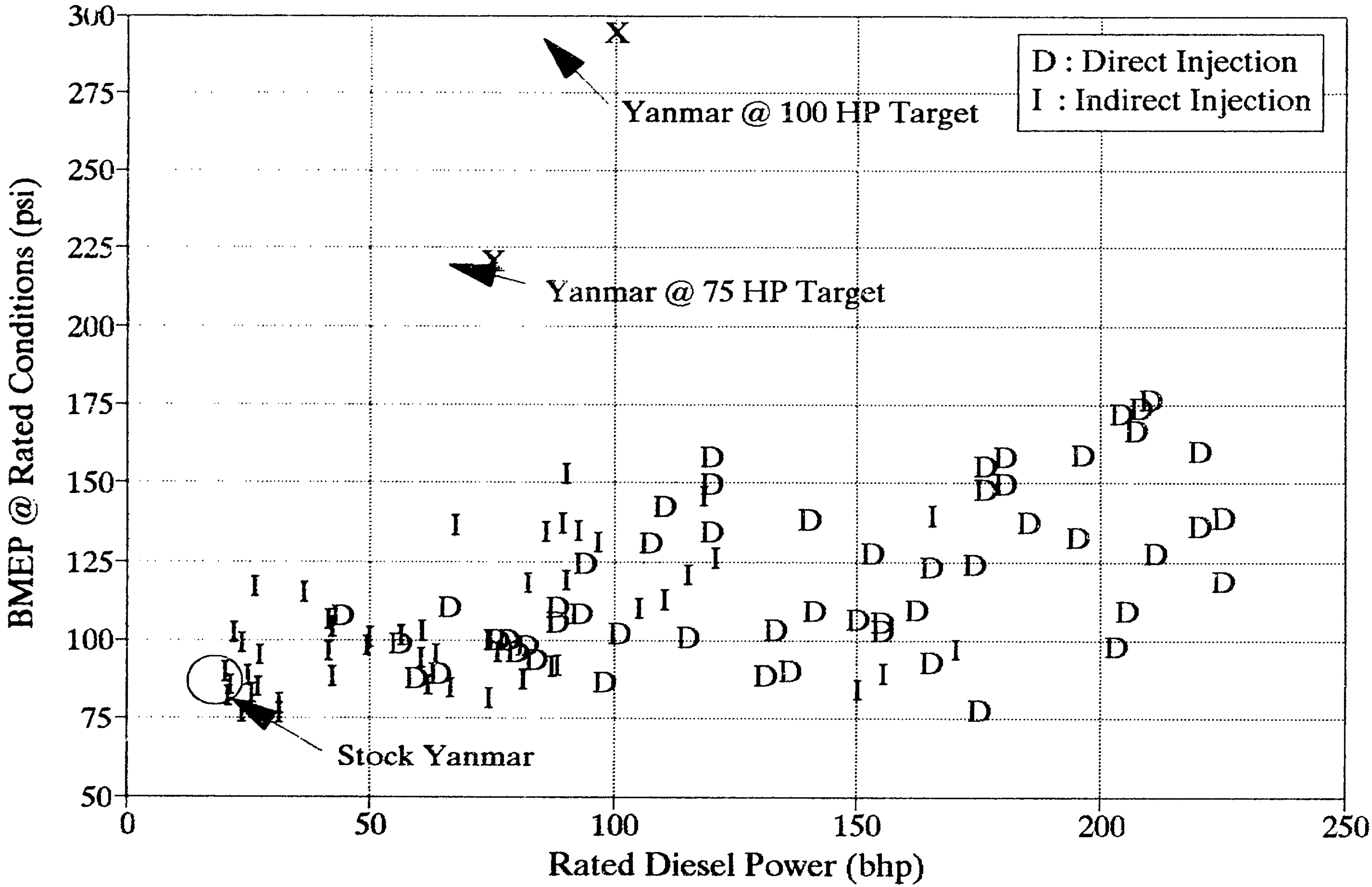
Perhaps more importantly than the structural loads, the thermal loading due to the high power density will require significant engine (data derived from Appendix) modifications. Fig. 22 gives the thermal loading for a wide range of diesel production engines and shows that the horsepower targets will require the Yanmar engine to exceed most commercial engine levels. However, the projected thermal loading values for the Yanmar are representative of other military engines, which are either currently in production, or are being marketed to the military (see Fig. 22). The high thermal loading can be addressed by aggressive piston cooling with oil jets and insulated exhaust port liners. These are both proven technologies that are in production on various engines.

In summary, the Yanmar engine could be modified to produce the required power and fit into the available space. Engine speed should not be a major problem, and mechanical and thermal loading can probably be addressed without excessive cost or technical risk. Although the APU weight goals will not be met, the improved specific fuel consumption compared with a turbine engine reduces the fuel weight required for a given mission duration. Engine life will be the primary unknown, and engine durability tests will be required to determine the exact effects of increasing the speed, power, and BMEP as required.

2. Convert Gasoline Engine to Spark-Assisted Diesel Operation

Another possibility for meeting the project targets is to convert an existing gasoline engine to spark-assisted diesel (SPAD) operation. A gasoline engine is chosen for this approach due to its low weight. A two-stroke engine may be chosen due to its small package volume in the absence

Diesel BMEP Trends



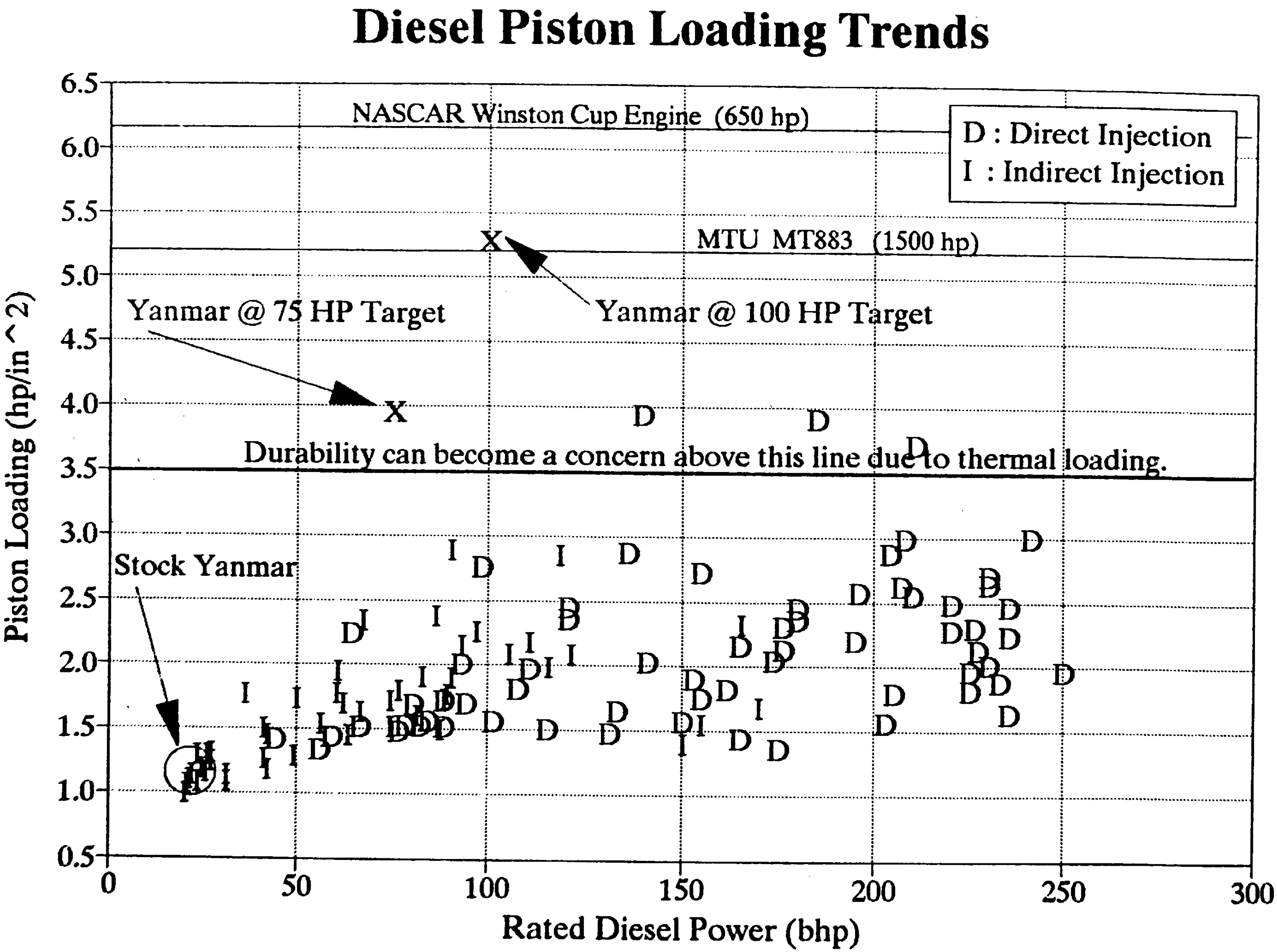


Figure 22. Thermal engine loading characteristics for a wide range of small diesel engines

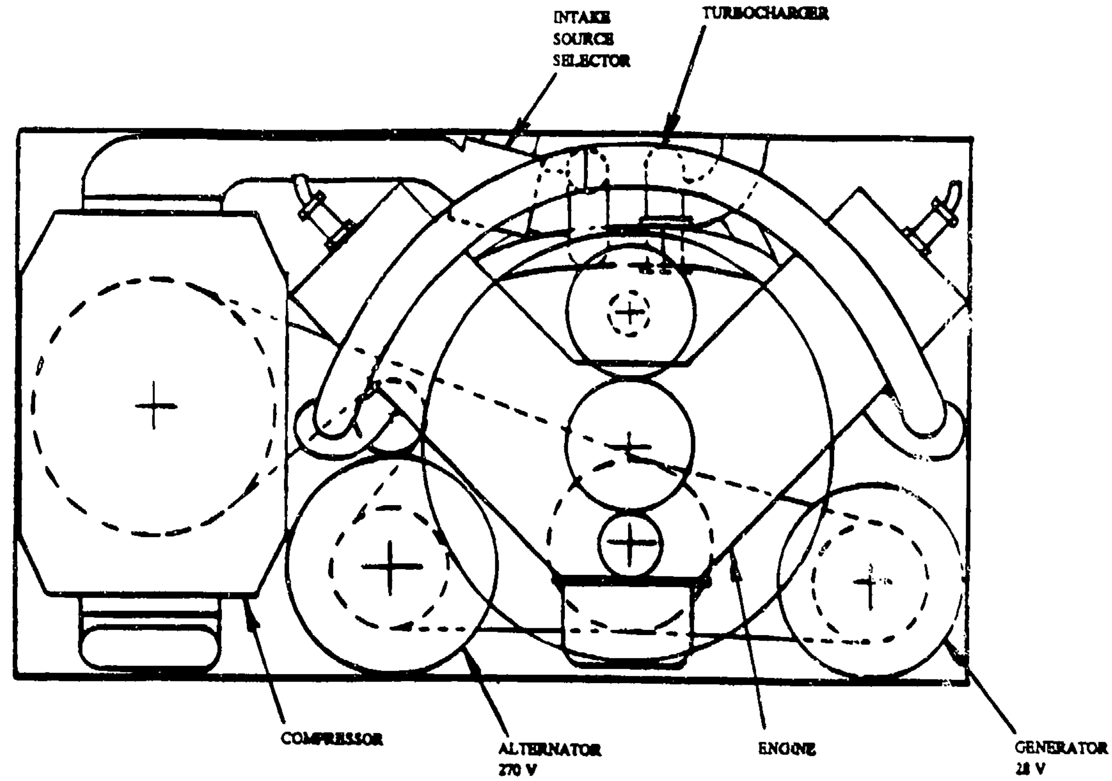
of any valve train. However, the low weight and strength of the gasoline engine eliminates the possibility of running at high compression ratios for compression ignition. Therefore, the spark-assisted diesel combustion process is necessary to operate reliably on diesel fuel.

SwRI has converted two-stroke gasoline engines to the SPAD combustion process with some success, although none of these engines have gone into full production. A 115-hp Johnson outboard engine appears to be a good match for the requirements of this application due to its displacement and overall dimensions. General specifications for the gasoline version of this engine are shown in TABLE 5.

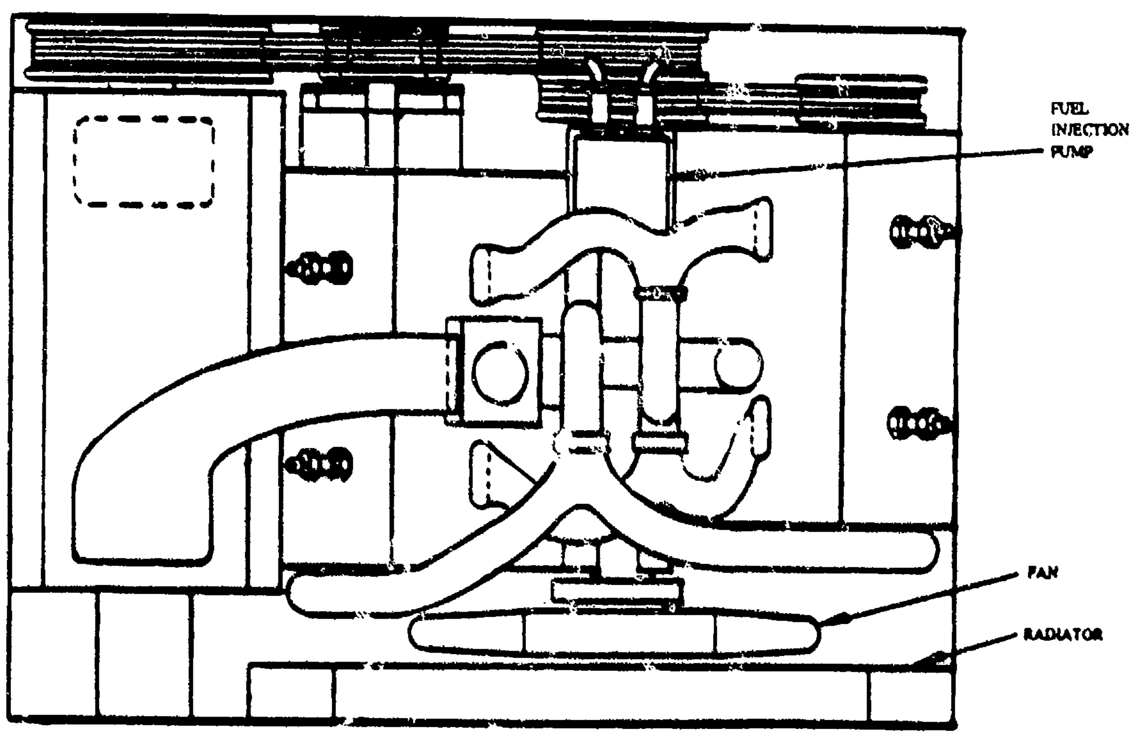
TABLE 5. Johnson Outboard Engine Specifications

Manufacturer:	Johnson
Model:	115-hp
Displacement:	1632 cc (99.6 cu in)
Rated Power:	115-hp at 4500 RPM
BMEP:	101 psi at 115-hp
Configuration:	V-type, four-cylinder
Cycle:	Two-Stroke
Induction:	Piston Ported, Crankcase Scavenged
Combustion:	Carbureted Open Chamber
L x W x H:	17 x 22 x 24 in.
Ignition:	Spark
Fuel Delivery:	Carbureted

Fig. 23 illustrates a possible packaging arrangement with this engine. This figure illustrates the advantage of a piston-ported, two-stroke, V-type engine with regards to fitting over 1.6L displacement within the package volume compared with the 0.9L four-stroke, in-line engine. Since the two-stroke engine already produces 115-hp at 4500 RPM, it is reasonable to expect to produce 75-hp with the SPAD system without turbocharging. 100-hp will probably require turbocharging for this engine. If the diesel version of this engine is rated at 5000 RPM, the BMEP requirements will be 60 psi and 80 psi for 75- and 100-hp, respectively. Thus, the mechanical loading on the two-stroke engine should not be a major factor.



END VIEW



TOP VIEW

Figure 23. Illustration of APU configuration using the Johnson outboard engine

The fuel injection equipment will be a significant development requirement for this engine if direct injection is adopted. Although acceptable DI fuel systems exist for four-stroke engines, the two-stroke engine requires approximately half the injection quantity per stroke, twice as often as the four-stroke engine. The expected difficulties with a DI fuel system suggest the application of an IDI combustion system with the two-stroke engine. An added benefit of this combustion system is the improved air utilization due to the limited amount of fuel and air mixing in the two-stroke cylinder.

The turbocharger will help reduce exhaust noise and temperature on the 100-hp model and may be desired for the 75-hp version for strategic purposes. A further advantage of turbocharging is the possibility of scavenging the engine externally and using a wet sump with journal bearings. The roller bearings typically used on outboard engines decrease engine life, perhaps below the targets for this project. The Johnson engine is liquid cooled, and oil could be used for the cooling media if integration with the transmission heat exchangers is acceptable.

3. Lightweight Diesel Engine Design and Development

Several new engine and/or compressor designs could be considered for this application. New engine designs consist of more compact V-type engines (i.e., 15-degree V instead of 60- to 90-degree V), integral engines and compressors such as those used in the pipeline industry, and new compressor designs such as the Vairex or other novel compressors. If a new engine design is considered, it should be designed from a "clean sheet" as a lightweight diesel engine, regardless of the configuration or cycle.

Extensive finite element analysis (FEA) should be conducted on all load bearing members of the engine, and special attention should be paid to the heat rejection characteristics of the engine. SwRI designed a lightweight diesel engine during the late 1970's that showed great promise for meeting similar goals as this project. Unfortunately, funding for the project was discontinued before a prototype could be built and tested.

Volkswagen has recently announced production plans for a 15-degree V passenger car engine. The advantage of such a narrow bank angle is the use of only one cylinder head, and a more compact engine package. This type engine configuration could provide a slight improvement in power to package volume ratio compared with a conventional V engine.

Assuming that a four-cylinder engine would be capable of providing the required power output for the application, a multi-cylinder, V-type engine that used additional cylinders to supply compressed air could reduce the total package volume for the engine and compressor. This proven technology has been successfully used in the gas pipeline and other industries for many years. A feasibility study should be conducted on this approach before launching a demonstration project on this concept.

During the course of this project, the compressor size was identified as a major obstacle facing this project. Numerous manufacturers were contacted, and technical specifications were obtained for their compressors that would supply the air flow and pressure ratios needed. Centrifugal, axial flow, rotary vane, and turbine pumps were investigated, and each was determined to have disadvantages for this application. One pump that may deserve more investigation is the Vairex pump. This pump is a positive displacement blower with variable stroke capabilities.

D. Cost Comparison

Cost information was obtained from a turbine engine manufacturer* regarding purchase costs and development costs of a turbine power plant that would meet the size and power requirements of this project. These numbers are compared with projected development and purchase costs for the other three alternatives described in this report in TABLE 6.

*Sundstrand Corporation

TABLE 6. Estimated Cost Comparisons

Engine Model and Description	Purchase Cost	Development Cost
Sundstrand T-150[1] (Turbine APU currently under development for 1994 army LH helicopter. Available mid 1994)	$60,000	$500,000
Sundstrand T62T-40C[1] (Modified production "Titan" APU. Available mid 1993)	$55,000	$1.5M
Yanmar 3TNA72E (Based on 600 units/yr, 10 year contract)	$1688[2] $10,000[3]	$750,000
Johnson 115-hp	$5,000[4] $12,000[5]	$1M
New Lightweight Diesel	$15,000 - 20,000	$2 - 3M

1 - Quote supplied to SwRI by Sundstrand dated September 3, 1991
2 - Current production 21-hp version
3 - Modified 75/100-hp estimate with premium grade components
4 - Current production gasoline 115-hp version
5 - Modified spark-assisted diesel estimate

This table illustrates that even if the Army spent $3M to develop a new lightweight diesel engine, the increased development costs compared with the T62T-40C would be offset by lower engine costs, and a cost incentive would exist for the diesel engine if more than 42 engines are purchased.

E. Technology Assessment

From a strictly technological standpoint, the "clean sheet" engine design approach has the most promise for meeting all the project goals simultaneously. However, even with a "clean sheet" design, the BSFC target for the 100-hp engine will be almost impossible to meet while satisfying the other engine specifications.

The second most attractive technical approach would be to increase the power of an existing diesel engine that will fit into the available space. The major challenges for this approach will be engine life and BSFC targets.

Finally, the conversion of an existing gasoline engine to spark-assisted diesel represents a similar cost alternative to increasing the power of a small diesel engine. However, potential difficulties associated with optimizing the spark-assisted diesel combustion system suggest that this approach will have a higher degree of technical risk than the small diesel engine upgrade.

The three approaches discussed above are prioritized in TABLE 7 according to cost, technical risk, and overall feasibility.

TABLE 7. APU Engine Concept Priorities

	Relative Development Cost	Relative Technical Risk	Overall Feasibility
New Lightweight Diesel Design	high	medium	low
Small Diesel Upgrade	low	low	high
Gasoline Conversion to Spark-Assisted Diesel	medium	high	medium

As TABLE 7 illustrates, the new lightweight diesel engine will be the most expensive to develop and will have a certain amount of technical risk. The small diesel engine modification will be the least expensive to develop and have the lowest technical risk. The gasoline engine conversion will also be less expensive to develop than the new lightweight diesel engine, but the challenge of optimizing the SPAD combustion system represents a substantial technical risk. Thus, increasing the power output of the small diesel engine appears to have the best overall feasibility. Even though it has relatively moderate technical risk, the new engine design probably has the lowest overall feasibility due to budget constraints.

F. Fuel Combustion Methodology

1. Load Conditions

Four-stroke engines should be able to maintain rated conditions for longer periods of time than two-stroke engines without dramatically compromising engine life due to thermal loading. This is because two-stroke engines will have a more difficult time rejecting heat than four-stroke engines. Two-stroke engines can be over-scavenged with an external blower in order to cool the combustion chamber. However, this method requires more power for scavenging, partially negating the benefit of running at high BMEP levels.

2. Engine Size

One of the greatest limitations on small diesel engines today is their fuel injection equipment. Acceptable DI fuel injection systems are commercially unavailable for small, high-output diesel engines. Although these systems are available for larger diesel engines, the smaller fuel quantities and higher operating speeds of small diesel engines makes it difficult to apply standard DI fuel system technology. Thus, an IDI combustion system or dramatically improved DI fuel injection system will probably be required for this application.

3. Engine Speed

Again, fuel injection systems are one of the limiting factors to increasing diesel engine speed. Fuel injection systems are available for 4500-5000 RPM DI four-stroke diesels, but not for high-speed two-stroke engines. A two-stroke diesel engine running at 5000 RPM will probably require an IDI combustion system. Similarly, a high-speed (i.e., 5000 RPM) four-stroke diesel will probably benefit from an IDI combustion system in terms of power density, but the BSFC will suffer compared with a DI system.

4. Response to Step Loading

Both compression ignition and SPAD combustion systems should respond well to step loading. In both cases, the engine will run unthrottled, so air will already be present in the combustion chamber. Increasing load will simply be a matter of adding fuel.

5. Starting and Operation Under Military Extreme

The SPAD could have advantages over the IDI or DI combustion systems under extremely cold conditions. The IDI and DI systems will require glow plugs for cold start operation. Hot starting should not be a problem for any of the engines. Turbo modifications may be required for high altitude conditions.

III. CONCLUSIONS

A number of conclusions were developed as a result of this study:

1. No commercially available diesel engine will meet the power and packaging goals without substantial modification and development.

2. A production gas turbine engine will meet the power, reliability, durability and packaging goals with only minor modifications. However, very few small gas turbine engines are available in this power range and each has notoriously poor fuel consumption and high purchase costs.

3. An existing small, production four-stroke diesel engine can be modified to increase the power output to meet most of the goals of this application. The engine weight and fuel consumption will be the most difficult targets to meet with this approach.

4. An existing, production two-stroke gasoline engine can be converted to operate on diesel fuel and meet many of the targets for this application. Difficulties associated with the development of a high-speed, two-stroke diesel combustion system are the primary drawbacks of this approach.

5. A dedicated, lightweight diesel engine can be designed and developed for this application that would include an integrated engine/compressor. This approach is expected to have the greatest potential for meeting all project goals simultaneously. Of course, the cost of this approach will be the highest compared with the previous approaches.

6. The fuel consumption level of the engine chosen for this application has a profound effect on the total weight and package volume required for the engine and fuel.

7. The turbine engine cannot compete with a diesel engine in terms of weight and package volume when the fuel storage volume and weight are taken into account.

8. The following design features are considered to be desirable for this engine application:

   (a) Liquid Cooling
   (b) Turbocharging with aftercooling
   (c) Indirect Injection (IDI) fuel system
   (d) V cylinder arrangement

IV. RECOMMENDATIONS

SwRI recommends that an existing diesel or gasoline engine be modified or converted to satisfy as many project goals as possible for demonstration purposes. The successful demonstration of an internal combustion in this application will emphasize the fuel economy improvements compared with a turbine engine.

After successfully demonstrating the concept of an IC engine in this application, a lightweight diesel engine should be designed and developed to meet the maximum number of project targets simultaneously.

LIST OF ACRONYMS AND ABBREVIATIONS

APECS	-	Auxiliary Powered Environmental Control System
APU	-	Auxiliary Power Unit
bhp	-	Brake Horsepower
BMEP	-	Brake Mean Effective Pressure
BSFC	-	Brake Specific Fuel Consumption
DI	-	Direct Injection
IC	-	Internal Combustion
IDI	-	Indirect Injection
NA	-	Naturally Aspirated
PSIG	-	Pounds Per Square Inch, Gauge
RPM	-	Revolutions Per Minute
SCFM	-	Standard Cubic Feet Per Minute
SPAD	-	Spark-Assisted Diesel
SwRI	-	Southwest Research Institute
T	-	Turbocharged
TA	-	Turbocharged With Aftercooling

APPENDIX

ENGINE DATA

The following tables contain data used to generate the plots for the Auxiliary Power Unit study. The following are calculated values:

- All specific output values: weight, volume, displacement, piston area
- Package volume
- Displacement
- Mean piston speed
- Fuel delivered per injection stroke
- BMEP

For the Cuyuna experimental engine: Power and specific outputs are targets. Dimensions are estimated for typical application and are subject to change:

Explanation of Codes

Combustion System:

D = Direct injection (DI)
I = Indirect injection (IDI)
S = Spark assist (SA), (direct injection)

Aspiration:

N = Naturally aspirated (NA)
T = Turbocharged (no aftercooling)
W = Turbocharged, Water-to-air aftercooled
A = Turbocharged, Air-to-air aftercooled

Application Code

A = Aircraft (Note: VM engine is not yet in production)

C = Commercial

M = Military

E = Experimental

Configuration (of cylinders)

I = Inline

V = Vee

H = Horizontally opposed (single crankshaft, not opposed piston)

O = Opposed piston (two crankshafts)

Note: The configuration of Rolls Royce K 60 was not known when the plots were made. The "O" on the plots is the VM aircraft diesel (supposedly "production-ready," but has not yet been FAA certified). The VM was later classified as "H" to distinguish horizontally opposed engines from opposed piston engines.

ENGINE MANUFACTURER	MODEL	INTERMIT POWER kW	RATED SPEED rpm	SPECIFIC OUTPUT			RATED POWER	ENG WGT kg	DIMENSIONS (cm) LENGTH	HEIGHT	WIDTH	[illegible]	ENG CYC (2/4)	COMBUST SYSTEM (D/I/S)	ASPIR	COOLING A/W	DISPL (l)	APPL	COUNTRY OF MFCR	MAX TORQUE	MAX TORQUE SPEED rpm	BSFC @ TORQUE PEAK	NO OF CYL	CONFIG (INLINE/VEE/OPPOSED)	BORE mm	STROKE (mm)	MEAN PISTON SPD @ RATED POWER	FUEL DELIV PER INJCTN	BMEP @ RATED POWER	PISTON AREA SPECIFIC POWER
LISTER PETTER	LPA2	14					0.411						4	D	N	A		C	UK				2	I						
KUBOTA	[illegible]	20	[illegible]	0.126		[illegible]		199					4	I	N	W	0.60	C	Japan	35	2600		4		64.0	62.2	1469		90	1.00
KUBOTA	D905B	21	3600	0.105		22.8		196					4	I	N	W	0.90	C	Japan	40	2000		3		72.0	73.6	1739		82	1.08
YANMAR	JTNA72	21	3600	0.100	5.63	23.9	0.429	209	15.5	19.6	17.3	3.73	4	I	N	W	0.88	C	Japan	38	1800	0.419	3	I	72.0	72.0	1701	14	86	1.11
LOMBARDINI	LDW903	22	3000	0.116	6.95	23.7		187	17.3	19.6	15.9	3.13	4	I	N	W	0.92	C	Italy	38	2200		3		72.0	75.0	1476		103	1.15
LISTER PETTER	LPA3	23	3600	0.061	5.24	21.4		267	20.4	20.9	18.1	1.45	4	I	N	A	1.09	C	UK				3		76.0	80.0	1890		77	1.11
SHANZI	DVA1030	23	3000	0.116	5.41	23.0		301	17.4	23.6	17.7	2.44	4	I		A	1.02	C	China?				2		86.0	88.0	1732		99	1.30
YANMAR	JTN75E	25	3600	0.090	4.76	24.7		271	20.8	22.2	19.3	5.15	4	I		W	0.99	C	Japan				3		75.0	75.0	1772		89	1.19
LOMBARDINI	[illegible]	26	3000	0.099	3.55	19.2		287	25.8	24.3	21.7	7.26	4	I	N	A	1.33	C	Italy	56	1800		2		95.0	94.0	1850		83	1.17
KUBOTA	V800TB	26	3600	0.157		12.5		165					4	I	T	W	0.80	C	Japan	44	2600		4		64.0	62.2	1469		117	1.30
LISTER PETTER	LPW4	27					0.395						4	D	N	W		C	UK			0.370	4	I						
YAMAHA	ME1100	27	5100	0.059	2.43	20.4		152	28.1	23.5	29.2	11.14	4	I	N	W	1.32	C	Japan				2		91.5	100.0	2031		85	1.32
LISTER PETTER	LPW4	27					0.452						4	I	N	W		C	UK			0.419	4	I						
KOMATSU	2D94	27	2500	0.063	3.96	18.1	0.434	430	19.9	26.5	20.7	6.74	4	I	N	W	1.47	C	Japan	65	1600	0.429	2	I	91.0	105.9	1737	45	95	1.26
LISTER-PETTER	LPW3	31	3600	0.095	6.93	22.2		315	22.1	20.9	16.7	4.47	4	I	N	W	1.39	C	UK				3		86.0	80.0	1890		80	1.15
LISTER-PETTER	LPA4	31	3600	0.090	5.83	21.3		344	24.3	20.9	18.1	5.31	4	I	N	A	1.45	C	UK				4		76.0	80.0	1890		77	1.10
YAMAHA	ME-1300T	36	5100	0.075	3.25	27.5		485	28.3	23.3	29.2	11.14	4	I	T	W	1.32	C	Japan				2		91.5	100.0	2034		115	1.78
ISUZU	QD-40	41	2800	0.056	4.29	23.0	0.452	478	28.1	28.4	20.7	9.66	4	I		W	1.78	C	Japan	80	2000	0.416	3	I	86.1	102.1	1876	36	107	1.51
KOMATSU	3D94	41	2500	0.080	4.75	18.6	0.484	514	26.6	29.3	20.7	8.61	4	I	N	W	2.20	C	Japan	97	1600	0.429	3	I	94.0	105.9	1737	46	97	1.27
ONAN	L317	42	3600	0.094	6.82	24.5	0.483	448	22.7	24.3	19.3	6.16	4	I	N	W	1.71	C	USA	73	2000	0.442	3	I	88.9	91.9	2172	32	88	1.46
KUBOTA	V1902B	42	2800	0.098	5.12	22.5	0.490	428	28.2	24.9	20.2	8.21	4	I	N	W	1.87	C	Japan	88	1600	0.430	4	I	85.1	82.0	1507	29	104	1.19
PERKINS	4.108	44	3000	0.086	6.17	24.9	0.455	510	25.4	25.0	19.4	7.33	4	D	N	W	1.77	C	UK	81	2300	0.423	4	I	79.5	88.9	1750	29	108	1.43
YAMAHA (Natl)	6FO	44	5500	0.393	27.25	55.6	0.459	113	16.0	22.0	8.0	1.63	2	S	N	W	0.80	E	Japan				2	I	84.0	72.0	2598	16	66	2.58
TOYOTA	2L2.5D	49	3600	0.093	5.21	19.8		529	25.5	27.4	23.3	9.43	4	I	N	W	2.48	C	Japan	100	2200		4		88.0	102.0	1740		99	1.30
VOLKSWAGEN	068.D	50	4800	0.175	6.83	31.2	0.485	284	30.5	24.1	25.4	7.26	4	I	N	W	1.59	C	Germany	70	2000	0.419	4	I	76.5	86.4	2268	26	101	1.74
CUYUNA		50	6000	1.000	8.55	91.9	0.400	50	16.5	16.1	33.9	5.86	2	S	N	W	0.54	E	US				2	I	73.0	65.0	2559	29	99	3.85
JOHN DEERE	3179D	56	2500	0.074	5.51	19.1	0.385	760	27.6	32.0	19.9	10.17	4	D	N	W	2.94	C	USA	143	1200		3	I	106.4	110.0	1804	50	99	1.35
ISUZU	QD-60	56	3000	0.114	5.36	23.6	0.470	491	31.5	27.3	21.0	10.45	4	I		W	2.38	C	Japan	108	2000	0.440	4	I	86.1	102.1	2010	38	102	1.55
PERKINS	4.2032	59	2600	0.114	4.76	17.7	0.433	516	12.1	29.4	22.7	12.40	4	D	N	W	3.34	C	UK	152	1500	0.420	4	I	91.4	127.0	2167	42	88	1.45
NISSAN	SD22	60	3600	0.127	6.77	27.7	0.527	474	27.2	24.6	22.9	6.87	4	I	N	W	2.17	C	Japan	96	1800		4	I	83.1	100.1	2195	36	95	1.79
VOLKSWAGEN	028.B	60	4800	0.210	8.26	31.8	0.485	267	30.5	25.0	24.6	7.30	4	I	N	W	1.90	C	Germany	92	2250	0.411	4	I	79.5	95.5	2507	32	103	1.96
RENAULT	J8S706	62	4500	0.175		29.8		353					4	I	N	W	2.07	C	France	91	2250		4		86.0	89.0	2628		86	1.71
KOMATSU	4D94-2	63	2800	0.099	5.55	21.4	0.475	639	30.0	29.3	22.4	11.39	4	I	N	W	2.94	C	Japan	132	1600	0.427	4	I	94.0	105.9	2015	45	96	1.47
DETROIT DIESEL	2-71	64	2000	0.067	3.12	27.5	0.460	900	32.0	41.0	27.0	30.50	2	D	N	W	2.32	C	USA	186	1200	0.468	2	I	108.0	127.0	1667	64	89	2.26
KUBOTA	D1302B	66	2400	0.095	5.68	20.5	0.390	695	27.9	33.0	21.8	11.62	4	D	N	W	3.22	C	Japan	165	1300	0.375	3	I	109.0	115.1	1812	62	111	1.52
DAIMLER-BENZ	OM616	66	4300	0.167	6.61	27.5	0.578	443	27.4	28.0	22.5	9.99	4	I	N	W	2.40	C	Germany	99	2400	0.442	4	I	90.9	92.5	2548	39	85	1.64
VOLKSWAGEN	068.C	67	4800	0.227	8.71	42.2	0.477	295	30.5	25.0	26.0	7.70	4	I	T	W	1.59	C	Germany	98	2200	0.452	4	I	76.5	86.4	2268	35	137	2.35
VOLKSWAGEN	075.1	74	4000	0.175	6.34	31.0	0.493	421	30.5	26.5	24.9	11.65	4	I	N	W	2.38	C	Germany	100	3000	0.444	6	I	76.5	86.4	2268	26	100	1.73
MAZDA	SL35	74	3400	0.129	6.61	21.4		573	29.1	28.1	23.5	11.14	4	I		W	3.46	C	Japan				4		100.0	110.0	2454		82	1.52
CUMMINS	4B3.9	76	2500	0.112	5.83	19.4	0.358	680	30.1	32.6	22.9	13.00	4	D	N	W	3.93	C	USA	184	1200	0.373	4	I	102.1	119.9	1967	51	101	1.50
FIAT	8140.61	76	4200	0.158	9.52	31.1	0.380	481	23.5	27.3	21.5	7.96	4	I		W	2.44	C	Italy	110	2400	0.410	4	I	93.0	89.9	2478	37	96	1.81
ISUZU	QD-90	78	2800	0.112	4.79	21.7	0.400	694	34.2	30.4	27.1	16.31	4	D		W	3.60	C	Japan	162	2000	0.370	4	I	102.1	110.0	2021	48	100	1.54
DAF	DF385	80	2800	0.112	4.65	20.8		717	32.9	34.3	26.2	17.11	4	D		W	3.86	C	Holland				4	I	98.3	127.0	2333		96	1.70
DAIMLER-BENZ	OM617	81	4000	0.146	5.56	27.0	0.539	556	34.1	28.0	26.4	14.59	4	I	N	W	3.00	C	Germany	125	2400	0.412	5	I	90.9	92.5	2427	38	88	1.61
KOMATSU	4D105-5	82	2500	0.093	4.25	19.0	0.366	882	36.2	35.3	26.1	19.30	4	D	N	W	4.32	C	Japan	197	1600	0.379	4	I	104.9	125.0	2050	55	99	1.53
PEUGEOT	XD3T2	82	3600	0.167		32.9		492					4	I	T	W	2.50	C	France	140	2000		4		94.0	90.0	2126		119	1.91
MWM	D 226-4	84	2800	0.107	6.95	20.5	0.414	783	33.4	31.1	20.1	12.06	4	D	N	W	4.14	C	Germany	177	1600		4	I	104.9	119.9	2203	54	94	1.57
RENAULT	J8S814	86	4000	0.229		41.5		375					4	I	T	W	2.07	C	France	135	2250		4		86.0	89.0	2336		135	2.38
ONAN	L634	87	3600	0.125	7.61	25.4	0.370	694	37.5	26.5	19.6	11.39	4	I	N	W	3.42	C	USA	193	2000	0.422	6	I	88.9	91.9	2172	33	92	1.51
NISSAN	SD33	88	3800	0.133	5.23	27.1	0.532	661	30.4	32.4	22.8	16.61	4	I	N	W	3.25	C	Japan	135	1800		6	I	83.1	100.1	2495	35	92	1.75
RENAULT	720	88	3000	0.129	7.53	24.4	0.412	683	31.3	31.7	20.5	11.70	4	D		W	3.60	C	France	168	1500		4	I	102.1	110.0	2165	52	106	1.73
KUBOTA	[illegible]	88	2400	0.105	6.20	20.5	0.390	841	32.7	31.4	21.6	14.19	4	D	N	W	4.29	C	Japan	222	1300	0.370	4	I	109.0	115.1	1812	62	111	1.52
HERCULES	D2300T	89	2400	0.124	6.78	25.4		717	32.4	30.2	23.2	13.13	4	I	T	W	3.50	C	USA	205	2000		4		101.6	108.0	1701		137	1.77
PEUGEOT	XUD7T	90	4300	0.272		50.9		331					4	I	A	W	1.77	C	France	133	2100		4		80.0	88.0	2483		153	2.89
NAVISTAR	DT-239	90	2500	0.097	5.83	23.0	0.433	925	28.9	44.0	21.0	15.45	4	I	N	W	3.92	C	USA	206	1600	0.416	4	I	98.6	128.5	2108	67	119	1.90
VM MOTORI	HR494HT	93	3500	0.182	8.59	33.3		507	28.7	27.0	24.1	10.76	4	I	T	W	2.78	C	Italy				4		94.0	100.0	2100		135	2.15
DAIMLER-BENZ	OM364	93	2800	0.126	6.01	23.4	0.376	739	27.9	35.0	27.4	15.48	4	D	N	W	3.98	C	Germany	200	1400		4	I	97.5	133.1	2445	54	108	2.01

ENGINE MANUFACTURER	MODEL	INTERMITTENT POWER (hp)	RATED SPEED (rpm)	SPECIFIC OUTPUT WGT (hp/lb)	SPECIFIC OUTPUT PKG VOL (hp/ft^3)	SPECIFIC OUTPUT DISPL (hp/L)	BSFC @ RATED POWER (lb/hp-hr)	DRY WGT (lb)	DIMENSIONS (in) LENGTH	DIMENSIONS (in) HEIGHT	DIMENSIONS (in) WIDTH	PKG VOL (ft^3)	ENG CYC (#)	COMBUST SYSTEM DI/IDI/SA	ASPIR	COOLING Air Water Oil	DISPL (L)	SPEC CODE	COUNTRY OF MFGR	MAX TORQUE (lb-ft)	MAX TORQUE SPEED (rpm)	BSFC @ TORQUE PEAK (lb per hp-hr)	NO OF CYL	CONFIG (INLINE, VEE, OPPOSED)	BORE (mm)	STROKE (mm)	MEAN PISTON SPD @ RATED POWER (ft/min)	FUEL DELIV PER INJCTN (mm^3/stroke)	BMEP @ RATED POWER (psi)	PISTON AREA SPECIFIC POWER (hp/in^2)
JOHN DEERE	4239T	94	2500	0.096	6.60	24.0	0.375	975	33.6	37.2	19.7	14.25	4	D	T	W	3.91	C	USA	233	1600	0.366	4	I	106.4	110.0	1804	61	125	1.71
VOLKSWAGEN	075 2	97	4000	0.223	7.90	40.5	0.473	432	30.5	26.5	26.2	12.23	4	I	T	W	2.38	C	Germany	140	2600	0.439	6	I	76.5	86.4	2268	33	131	2.26
DETROIT DIESEL	3-53N	98	2800	0.102	5.37	37.5	0.445	965	33.4	35.0	27.0	18.27	2	D	N	W	2.62	C	USA	205	1800	0.458	3	I	98.6	114.3	2100	45	87	2.76
FIAT	8340.04	101	2800	0.112	5.54	22.1	0.392	904	36.9	34.6	24.7	18.25	4	D		W	4.57	C	Italy	213	1600	0.352	4	I	115.1	110.0	2021	61	102	1.57
NISSAN	SD33T	105	3800	0.162	7.19	32.3	0.559	649	37.3	27.5	24.6	14.60	4	I	T	W	3.25	C	Japan	184	2000		6	I	83.1	100.1	2495	44	110	2.08
JOHN DEERE	4045T	107	2200			22.2	0.352						4	D	W	W	4.83	C	USA	310	1360	0.335	4	I	110.0	127.0	1833	74	131	1.82
VOLVO	TD 45B	110	2200			24.2	0.378		37.2	37.9			4	D	T	W	4.54	C	Sweden	313	1400		4	I	106.7	127.0	1833	82	143	1.99
DAIMLER-BENZ	OM617A	110	4300	0.199	7.27	36.7	0.500	553	34.1	26.0	27.4	15.14	4	I	T	W	3.00	C	Germany	172	2400	0.410	5	I	90.9	92.5	2548	45	113	2.19
ONAN	L634T	115	3600	0.157	6.71	33.6	0.467	734	37.5	32.4	24.4	17.16	4	I	T	W	3.42	C	USA	193	2000	0.422	6	I	88.9	91.9	2172	43	121	1.99
CUMMINS	6B5.9	115	2500	0.135	6.58	19.5	0.388	855	39.6	33.3	22.9	17.48	4	D	N	W	5.89	C	USA	270	1200	0.365	6	I	102.1	119.9	1967	51	101	1.51
VM MOTORI	HR 492 GU	115	4200	0.257	11.94	47.2		459	26.3	26.8	24.2	9.88	4	I	W	W	2.50	C	Italy				4		92.0	94.0	2591		146	2.86
CUMMINS	4BTA3.9	120	2500	0.166	8.20	30.6	0.365	725	30.1	36.7	22.9	14.64	4	D	W	W	3.93	C	USA	290	1700	0.360	4	I	102.1	119.9	1967	76	159	2.36
PERKINS	T4.40 CC	120	2600	0.154	6.97	30.0	0.365	778	32.9	33.9	26.7	17.23	4	D	T	W	4.00	C	UK	285	1600		4	I	100.1	127.0	2167	73	150	2.46
MITSUBISHI	4D31T	120	3500	0.185	8.34	36.4		650	32.1	25.7	27.0	14.39	4	D	T	W	3.30	C	Japan	209	1800		4	I	100.1	104.9	2409		135	2.46
CUMMINS (ONAN)	6AT2.1	121	3600			35.1	0.478						4	I	W	W	3.43	C	USA	236	1600	0.395	6	I	89.0	92.0	2173	46	127	2.09
NISSAN	ND6	131	2800	0.104	3.99	19.2	0.458	1257	47.0	42.4	26.5	32.87	4	D	N	W	6.83	C	Japan	297	1400		6	I	110.0	119.9	2203	62	89	1.48
STEYR	610	133	2800	0.112	5.97	22.3	0.408	1190	49.5	34.1	22.8	22.27	4	D		W	5.97	C	Austria	277	1600	0.380	6	I	104.9	115.1	2114	56	103	1.66
DETROIT DIESEL	4-53N	136	2800	0.123	6.07	39.0	0.428	1110	38.8	37.0	27.0	22.43	2	D	N	W	3.49	C	USA	282	1800	0.442	4	I	98.6	114.3	2100	45	90	2.88
DETROIT DIESEL	3-53T	140	2500	0.140	6.31	53.5	0.373	1000	33.4	40.3	28.5	22.20	2	D	T	W	2.62	C	USA	317	1600	0.384	3	I	98.6	114.3	1875	60	139	3.95
DAIMLER-BENZ	OM366	141	2800	0.144		23.6	0.374	981					4	D	N	W	5.97	C	Germany	303	1400		6	I	97.5	133.1	2445	54	110	2.03
CUMMINS	6C8.3	150	2200	0.119	5.54	18.1	0.368	1256	44.0	38.6	27.4	27.07	4	D	N	W	8.28	C	USA	400	1500	0.358	6	I	114.0	135.1	1951	72	107	1.58
CATERPILLAR	3306 NA	150	2200	0.073	4.39	14.4	0.430	2050	50.0	39.8	29.7	34.20	4	I	N	W	10.45	C	USA	427	1200	0.402	6	I	120.7	152.4	2200	84	85	1.41
MWM	D 226-6T	153	2500	0.165	7.50	24.6	0.405	1052	54.3	32.3	20.1	20.40	4	D	T	W	6.22	C	Germany	364	1600		6	I	104.9	119.9	1967	71	128	1.90
MITSUBISHI	6D14	155	2900	0.137	7.50	23.6		1135	47.3	32.7	23.1	20.68	4	D	N	W	6.56	C	Japan	300	1800		6	I	110.0	115.1	2190		106	1.76
DETROIT DIESEL	6.2 HD	155	3600	0.221	12.19	24.9	0.494	701	29.5	27.4	27.2	12.72	4	I	N	W	6.23	C	USA	250	2000	0.433	8	V	101.1	97.0	2292	46	90	1.56
DETROIT DIESEL	4-71N	155	2100	0.087	5.24	33.4	0.390	1780	42.0	42.0	29.0	29.60	2	D	N	W	4.65	C	USA	407	1600	0.404	4	I	108.0	127.0	1750	62	103	2.73
NISSAN	ND6T	162	2800	0.120	2.46	23.7	0.437	1356	53.0	65.1	47.6	65.84	4	D	T	W	6.83	C	Japan	353	1600		6	I	110.0	119.9	2203	73	110	1.83
DETROIT DIESEL	V8-8.2	165	2800	0.145	7.72	20.1	0.387	1139	35.0	32.7	32.3	21.39	4	D	N	W	8.20	C	USA	354	1200	0.376	8	V	108.0	112.0	2058	49	93	1.45
CATERPILLAR	3304T	165	2200	0.100	5.14	23.7	0.425	1655	37.6	65.7	32.3	32.12	4	I	T	W	6.97	C	USA	483	1400	0.410	4	I	120.7	152.4	2200	138	140	2.33
ISUZU	6BD1T	165	3000	0.156	6.39	28.5	0.405	1056	45.3	31.7	31.1	25.84	4	D	T	W	5.79	C	Japan	335	1900		6	I	102.1	117.9	2320	64	123	2.17
NAVISTAR	6.9 L	170	3300	0.200	7.33	24.7	0.470	850	34.4	33.0	35.1	23.19	4	I	N	W	6.89	C	USA	310	2000		8	V	101.6	106.2	2299	52	97	1.69
HINO	HO6C T	174	2900	0.145	6.54	26.8	0.390	1200	44.8	35.5	28.9	26.60	4	D	T	W	6.49	C	Japan	375	1600	0.355	6	I	108.0	118.1	2170	70	124	2.05
CATERPILLAR	3208 NA	175	2800	0.132	7.34	16.8	0.400	1325	37.5	33.8	32.5	23.84	4	D	N	W	10.42	C	USA	400	1400	0.360	8	I	114.3	127.0	2333	54	78	1.38
JOHN DEERE	6359A	176	2500	0.129	8.75	30.0	0.378	1365	43.6	40.7	19.6	20.13	4	D	W	W	5.87	C	USA	434	1200		6	I	106.4	110.0	1804	77	156	2.13
RENAULT	060212	176	2900	0.155	10.69	32.0	0.388	1135	42.1	26.0	26.0	16.47	4	D		W	5.50	C	France	405	1800		6	I	102.1	112.0	2058	70	148	2.31
CUMMINS	6BTA5.9	180	2500	0.199	9.45	30.6	0.361	905	39.5	36.4	22.9	19.05	4	D	W	W	5.89	C	USA	445	1700	0.339	6	I	102.1	119.9	1967	75	159	2.36
PERKINS	T6.60 CC	180	2600	0.157	8.45	30.0	0.365	1150	40.7	33.9	26.7	21.32	4	D	T	W	5.99	C	UK	406	1600		6	I	100.1	127.0	2167	73	150	2.46
DETROIT DIESEL	4-53T	185	2500	0.147	7.26	53.1	0.375	1260	38.8	38.1	29.8	25.49	2	D	T	W	3.49	C	USA	435	1500	0.390	4	I	98.6	114.3	1875	60	138	3.91
MITSUBISHI	6D14T	195	2900	0.162	7.71	29.7		1201	47.3	34.5	26.8	25.31	4	D	T	W	6.56	C	Japan	420	1400		6	I	110.0	115.1	2190		133	2.21
M.A.N.	DO226MKF	196	2800	0.207	7.54	34.4	0.370	948	45.0	34.1	29.3	26.02	4	D	W	W	5.70	C	Germany	461	1600		6	I	102.1	116.1	2133	75	159	2.57
NISSAN	PE6	203	2300	0.111	0.41	17.4		1830	53.2	47.6	341.6	500.60	4	D	N	W	11.68	C	Japan	533	1200		6	I	133.1	140.0	2112		98	1.57
VOLVO	TD 61 F	204	2800	0.159	6.01	37.2	0.365	1280	45.6	46.3	27.8	33.97	4	D	W	W	5.49	C	Sweden	457	1600		6	I	98.6	119.9	2203	77	172	2.88
KOMATSU	6D125-1	205	2200	0.110	5.99	18.6	0.355	1863	53.2	41.1	27.5	34.80	4	D	N	W	11.05	C	Japan	600	1400	0.347	6	I	125.0	150.1	2167	95	110	1.80
DAF	DT 620	207	2400			33.5			44.6	0.0	24.0	0.00	4	D	T	W	6.19	C	Holland	490	1600	0.345	6	I	103.9	121.7	2076		167	2.63
DAIMLER-BENZ	OM366LA	208	2600	0.191		34.9	0.348	1091	38.2				4	D	W	W	5.97	C	Germany	482	1400		6	I	97.5	133.1	2271	80	174	2.99
VM MOTORI	TPJ-1304 HF	210	2640	0.402	13.32	36.0	0.377	418	34.3	21.7	36.6	15.76	4	D	A	W	5.84	A	Italy				4	H	130.0	110.0	1906	129	177	2.55
DETROIT DIESEL	4-71T	211	2300	0.115	5.95	45.4	0.357	1830	44.4	44.2	31.1	35.32	2	D	T	W	4.65	C	USA	533	1400	0.363	4	I	108.0	127.0	1917	71	128	3.72
KOMATSU	SA6D110-1	220	2500	0.166	6.60	30.9	0.353	1323	47.3	41.6	29.3	33.36	4	D	W	W	7.12	C	Japan	557	1800	0.338	6	I	110.0	125.0	2050	89	160	2.49
ISUZU	6SA1T	220	2500	0.177	7.73	26.3	0.351	1245	48.7	38.0	26.6	28.49	4	D	T	W	8.38	C	Japan	390	1700	0.372	6	I	114.8	134.9	2213	89	136	2.29
MITSUBISHI	6D22	225	2200	0.114	3.12	20.2		1973	61.2	42.0	48.5	72.14	4	D	N	W	11.15	C	Japan	564	1400		6	I	130.0	140.0	2020		119	1.82
CUMMINS	LT-10-C	225	2100	0.116	5.98	22.5	0.340	1945	51.5	39.7	31.8	37.63	4	D	T	W	10.00	C	USA	675	1300	0.338	6	I	125.0	135.9	1872	105	139	1.97
JOHN DEERE	6466A	226	2100	0.126	7.71	29.6	0.355	1800	47.4	46.6	22.0	29.33	4	D	W	W	7.63	C	USA	735	1450	0.355	6	I	115.8	120.7	1662	110	183	2.31
GARDNER	6LXCT	227	1900	0.120		21.7	0.337	1892	60.5	47.5	0.0	0.00	4	D	T	W	10.45	C	UK	664	1400		6	I	120.7	152.4	1900	116	148	2.14
DEERE	8955A	228	2100			14.6							4				15.64	M	USA				8		140.0	127.0	1750		90	1.19
DETROIT DIESEL	V8-8.2T	230	2800	0.198	10.76	28.1	0.391	1162	35.0	32.7	32.3	21.39	4	D	T	W	8.20	C	USA	510	1700	0.360	8	V	108.0	112.0	2058	69	130	2.03
NAVISTAR	DTI-466	230	2600	0.156	6.83	30.1		1475	47.6	42.2	29.0	33.71	4	D	W	W	7.64	C	USA				6	I	109.2	135.9	2318		150	2.64
DETROIT DIESEL	6-71N	230	2100	0.107	4.89	33.0		2150	56.0	44.0	33.0	47.06	2	D	N	W	6.97	C	USA	611	1600	0.418	6	I	108.0	127.0	1750	0	102	2.70

ENGINE MANUFACTURER	MODEL	INTERMIT POWER (hp)	RATED SPEED (rpm)	SPECIFIC OUTPUT WGT (hp/lb)	SPECIFIC OUTPUT PKG VOL (hp/ft^3)	SPECIFIC OUTPUT DISPL (hp/l)	BSFC @ RATED POWER (lb per hp hr)	DRY WGT (lb)	DIMENSIONS (in) LENGTH	HEIGHT	WIDTH	PKG VOL (ft^3)	ENG CYC (4,2)	COMBUST SYSTEM (DI/IDI)	ASPIR	COOLING (Air Water Oil)	DISPL (l)	APPL CODE	COUNTRY OF MFGR	MAX TORQUE (lb-ft)	MAX TORQUE SPEED (rpm)	BSFC @ TORQUE PEAK (lb per hp hr)	NO OF CYL	CONFIG (INLINE, VEE, OPPOSED)	BORE (mm)	STROKE (mm)	MEAN PISTON SPD @ RATED POWER (ft/min)	FUEL DELIV PER INJCTN (mm^3/stroke)	BMEP @ RATED POWER (psi)	PISTON AREA SPECIFIC POWER (hp/in^2)
DAF	DKA 1160	233	2200	0.119	5.92	20.0		1962	54.7	45.4	27.4	39.36	4	D		W	11.64	C	Holland	582	1300		6	I	130.0	146.1	2108		118	1.89
CUMMINS	N-855-C BC1	235	2100	0.091	4.60	16.8	0.366	2590	60.7	51.2	28.4	51.08	4	D	N	W	14.02	C	USA	647	1500	0.388	6	I	139.7	152.4	2100	128	104	1.65
RENAULT	0620305	235	2400	0.122	6.26	26.7	0.399	1925	49.3	44.2	29.8	37.58	4	D		W	8.81	C	France	655	1400		6	I	119.9	130.0	2048	112	144	2.24
CUMMINS	6TCAU	235	2300	0.176	8.30	28.4	0.350	1335	44.0	40.6	27.4	22.33	4	D	W	W	8.28	C	USA	645	1500	0.338	6	I	114.0	135.1	1951	108	167	2.47
ROLLS ROYCE	K 60	240	2400	0.150	12.77	36.5		1600	40.0	26.0	29.0	18.80	2				6.58	M	UK				12	O	87.4	91.4	1440		99	2.15
VOLVO	TD 71FS	241	2400	0.161	8.36	35.8	0.362	1500	47.2	46.3	22.8	28.83	4	D	W	W	6.74	C	Sweden	649	1400		6	I	104.9	130.0	2048	105	193	3.00
CATERPILLAR	3208 DIT	250	2600	0.175	9.24	24.0	0.378	1400	36.2	35.1	36.8	27.06	4	D	T	W	10.42	C	USA	640	1400	0.356	8	V	114.3	127.0	2167	78	120	1.97

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